HEATING AND VENTILATION

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AND
VENTILATION

BY

JOHN R. ALLEN
DEAN OF THE DEPARTMENT OF ENGINEERING AND ARCHITECTURE
UNIVERSITY OF MINNESOTA; MEMBER AMERICAN SOCIETY
OF HEATING AND VENTILATING ENGINEERS; MEMBER
AMERICAN SOCIETY OF MECHANICAL ENGINEERS

AND

J. H. WALKER
SUPERINTENDENT OF CENTRAL HEATING, THE DETROIT EDISON
COMPANY ; MEMBER AMERICAN SOCIETY OF HEATING
AND VENTILATING ENGINEERS

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PREFACE

This book is offered as a text-book upon the subject of heating and ventilation for use in the engineering and architectural schools. It is also believed that the development of working methods of design and the including of the various tables and charts make the book of some value as a handbook for the practicing engineer and architect.

Calculus has been employed to some extent in the development of certain expressions, this having been deemed desirable for the sake of completeness. For architectural students and others not equipped with higher mathematics, such parts may be omitted, however, without destroying the structure of the book. Problems have been included at the end of many of the chapters in order to illustrate the principles involved, but it is felt that they can be profitably supplemented by the actual designing by the student of complete heating and ventilating systems for representative buildings of various types.

Acknowledgment is made to the American Blower Company and the Buffalo Forge Company for the use of various charts and tables.

Information as to the typographical errors which are doubtless present in this initial edition will be gratefully received.

J. R. A.
J. H. W.

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HEATING AND VENTILATION

CHAPTER I

HEAT

1. Heat.—Heat has long been known to be a form of energy. Modern theories as to the exact nature of heat conceive it to be a motion or agitation of the molecules, or extremely small particles, of which every body is composed. The intensity of the heat in a body is believed to be dependent upon the violence of this molecular disturbance. Every substance on the earth contains some heat and to say that a body is "cold," means simply that it contains a relatively small amount of molecular motion.

Heat and many other forms of energy are mutually convertible. For example, heat energy is converted into electrical energy in a generating plant and electric energy is re-converted into heat energy in an electric stove. Heat energy is converted into mechanical energy in a steam locomotive and some of this mechanical energy is re-converted into heat energy by the fric- of the locomotive brakes.

2. Measurement of Heat.—In measuring heat there are two quantities to be considered: the *intensity* of heat and the *amount* of heat. A small piece of white-hot metal may not contain as great a quantity of heat as a pail of warm water, but the intensity of the heat in the former is much greater. The intensity of heat is expressed by the word **temperature.** The temperature of a body is most easily measured by noting its effect upon some other substance.

One measure of the intensity of heat in a body is its ability to transmit heat to a body of lower temperature. Heat will flow from a body of higher temperature to one of lower temperature *but will never flow,* of itself, *from one body into another of higher temperature.* When two bodies of different temperatures are placed in thermal contact a heat exchange takes place until the two bodies are at the same temperature and thermal equilibrium

is reached. We may, therefore, state that two bodies are at the same temperature when there is no tendency for heat to flow from the one to the other.

3. Measurement of Temperature.—The measurement of temperature is usually based upon some arbitrary scale which is standardized by comparison with some well-established physical "fixed points." In mechanical engineering most measurements of temperature are made on the **Fahrenheit scale.** On this scale the freezing point of water is taken at 32° and the boiling point at sea level barometer at 212,° the tube of the thermometer between these points being divided into 180 equal parts or degrees. There is, however, an increasing use of the **Centigrade scale** among engineers. In the Centigrade scale the distance between the freezing point and the boiling point is divided into 100 equal parts. The freezing point on the scale is marked 0 and the boiling point is marked 100°. Both the Fahrenheit and the Centigrade scales assume an arbitrary point for the zero of the scale.

If the temperature Fahrenheit is denoted by t_f and the temperature Centigrade by t_c, then the *conversion from one scale to the other* may be made by the following equations:

$$t_f = \frac{9}{5} t_c + 32$$

$$t_c = \frac{5}{9} (t_f - 32)$$

The most common instrument for measuring temperature is the mercury thermometer. Mercury like most other substances undergoes an increase in volume when heated, and is particularly useful because the amount of its expansion for equal increments in temperature is nearly constant over a wide range in temperature. The thermometer is a glass tube of very fine bore with a bulb blown on one end and filled with mercury, as shown in Fig. 1. The air is expelled from the tube by boiling the mercury and the tube is sealed. The space above the mercury then contains mercury vapor at a very low pressure. The 32° and the 212° points of the Fahrenheit scale are located on the tube or stem by immersing the bulb in a freezing mixture and in boiling water. The distance between these points is then divided into 180 equal parts.

To do accurate work with the thermometer is much more difficult than is generally supposed. The mercury of the ordinary glass thermometer does not expand in exactly equal amounts for equal increments of temperature and the bore of the thermometer is never absolutely uniform throughout the length of the tube. All of these irregularities produce errors in observation. When measuring the temperature of liquids the depth to which the thermometer is immersed affects the reading and the thermometer should be calibrated at the depth at which it is to be used.

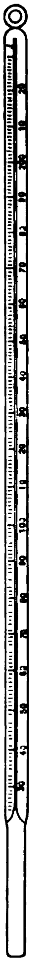


Fig. 1.

It is really its own temperature that the thermometer indicates and the accuracy with which the temperature of a substance is measured depends upon the completeness with which its temperature is reached by the thermometer. The thermometer must therefore be brought into intimate thermal contact with the substance to be measured. In measuring the temperature of fluids in pipes, a brass or steel well is inserted into the pipe and filled with some liquid such as oil or mercury, in which the thermometer is immersed. If the thermometer is used to measure the temperature of the air in the room in which there are objects of a higher temperature than the thermometer, its bulb must be protected from the radiant heat of these hot bodies; otherwise the thermometer will not read the temperature of the air surrounding it but will be affected by the radiant heat absorbed by it. When accurate temperature measurements are desired a careful study should be made of the thermometer and its errors and all inaccuracies should be allowed for by careful calibration.

The mercury thermometer can be used up to temperatures of 500°F. and for temperatures as low as −40°. Where lower temperatures must be measured it is customary to use thermometers filled with alcohol, and for temperatures higher than 500°F. some form of pyrometer must be used. High temperatures may be determined approximately by color. For each temperature there is a corresponding color and an approximation to the actual temperature can be made by an observation of the color of the heated substance. Table I gives the temperature colors.

TABLE I.—TEMPERATURE COLORS

Color	Temp. C.	Temp. F.
Faint red	525	977
Dark red	700	1,292
Faint cherry	800	1,472
Cherry	900	1,652
Bright cherry	1,000	1,832
Dark orange	1,100	2,012
Bright orange	1,200	2,192
White heat	1,300	2,372
Bright white	1,400	2,552
Dazzling white	1,500–1,600	2,732–2,912

4. Absolute Temperature.—In any theoretical consideration of heat it is necessary to have some **absolute scale** of temperature. The point at which the **molecules of a substance would have no motion** is considered to be the **absolute zero** point. According to Marks and Davis this point is theoretically at 491.64° below the freezing point of water on the Fahrenheit scale, or 459.64° below the Fahrenheit zero. On the Centigrade scale the absolute zero is at −273.1°. To convert any temperature on the Fahrenheit or Centigrade scale to absolute temperature the following formulæ are used:

$$T_f = t_f + 460 \quad \text{(approximately)}$$
$$T_c = t_c + 273 \quad \text{(approximately)}$$

in which the absolute temperatures on the Fahrenheit and Centigrade scales are represented by T_f and T_c. These expressions are sufficiently accurate for ordinary work.

No one has as yet been able to produce a temperature as low as the absolute zero. The lowest temperatures ever attained have been produced in the heat laboratory at Leyden, Holland, at which there has been produced a temperature of 489° below the Fahrenheit freezing point.

5. Unit of Heat.—Heat must be measured by the **effect which it produces** upon some substance. The **unit of heat** used in mechanical engineering is **the heat required to raise the temperature of a pound of water one degree Fahrenheit.** This is called the **British thermal unit** and is denoted by **B.t.u.** As this quantity

is not exactly the same at all temperatures it is necessary to specify further a definite temperature at which the unit is to be established. The practice of different authorities varies in this regard, but the **mean B.t.u.** established by Marks and Davis is becoming generally used. This is defined as the one hundred and eightieth part of the heat necessary to raise the temperature of one pound of water from 32° to 212°F.

6. Specific Heat.—**Specific heat** may be defined as **the heat necessary to raise the temperature of a unit weight of a substance through one degree.** It represents the **specific thermal capacity** of a body. In English units the specific heat is the quantity of heat necessary to raise a pound of a substance one degree Fahrenheit, expressed in British thermal units. Since the British thermal unit is the quantity of heat necessary to raise a pound of water one degree Fahrenheit, we may say that the specific heat represents the **ratio** between the heat necessary to raise a unit weight of a body one degree and the heat necessary to raise the same weight of water one degree.

When a substance is heated at **constant pressure** its volume increases against that pressure and **external work is done** as a consequence. The external work may be computed by multiplying the pressure by the change in volume. When heated at constant volume no external work is done as no movement is made against an external resistance. In any substance, such as a gas, which has a large coefficient of expansion due to heat, it is therefore necessary to distinguish between the **two specific heats,** the specific heat of **constant pressure** and the specific heat of **constant volume.** The difference between the two specific heats in any particular gas must be equal to the heat equivalent of the external work done when a unit weight of the gas is raised one degree at a constant pressure.

The quantity of heat added to or removed from a body is equal to

$$WC(t_2 - t_1)$$

in which

W = weight of the body in pounds.
C = specific heat of the material.
t_1 = lower temperature Fahrenheit.
t_2 = higher temperature Fahrenheit.

TABLE II.—SPECIFIC HEATS

Substance	Specific heat
Liquids:	
Water	1.0000
Alcohol	0.6220
Turpentine	0.4720
Petroleum	0.4340
Olive oil	0.3090
Metals:	
Cast iron	0.1298
Wrought iron	0.1138
Soft steel	0.1165
Copper	0.0951
Brass	0.0939
Tin	0.0569
Lead	0.0314
Aluminum	0.2185
Zinc	0.0953
Minerals:	
Coal	0.2777
Marble	0.2159
Chalk	0.2149
Stones generally	0.2100
Limestone	0.2170
Building Materials:	
Brickwork	0.1950
Masonry	0.2000
Plaster	0.2000
Pine wood	0.4670
Oak wood	0.5700
Birch	0.4800
Glass	0.1977

SPECIFIC HEAT OF GASES

Substance	Constant pressure	Constant volume
Air	0.2415	0.1729
Oxygen	0.2175	0.1550
Hydrogen	3.4090	2.4122
Nitrogen	0.2438	0.1727
Steam	0.5000	0.3500
Carbonic acid, CO_2	0.2479	0.1758
Ammonia	0.5080	0.2990

Example.—It is required to raise the temperature of a cast-iron radiator weighing 300 pounds from 70° to 212°. The temperature through which the iron would be raised would then be 212° minus 70° or 142°. From Table

II we see that to raise 1 pound of cast iron 1° would require 0.1298 heat units. To raise 1 pound 142° would require 142 times 0.1298 or 18.43 heat units, and to raise 300 pounds 1° would require 300 times this amount or 5529 B.t.u., the heat required to heat the radiator.

Example.—A church 80 by 100 feet inside has stone walls 2½ feet thick for 10 feet above the ground and for the remaining 20 feet 2 feet thick. The roof has a ½ pitch and is made of 2 by 8-inch rafters, 16 inches on centers, covered with 1 inch of pine boarding, tar paper and slate ¼ inch thick. Main floor composed of two 1-inch thicknesses of boards laid on 2 by 12-inch joists, 16-inch centers. Ceiling is of plaster ¾ inch thick. The church has 20 windows, 6 feet wide and 15 feet high, 12 windows 4 feet wide and 6 feet high, and 2 doors, 8 feet wide and 12 feet high. Allowing an addition of 15 per cent. for furnishings, find the heat required to raise the temperature of the structure from 0° to 50°.

Weight of stonework, stone weighing 160 pounds per cubic foot:

$$
\begin{array}{ll}
370 \times 10 \times 2\frac{1}{2} & = 9{,}250 \text{ cubic feet} \\
368 \times 20 \times 2 & = 14{,}720 \text{ cubic feet} \\
84 + 2 \times 40 \times 2 \times 2 & = 6{,}720 \text{ cubic feet} \\
\hline
 & \quad 30{,}690 \text{ cubic feet}
\end{array}
$$

Deduction for windows and doors:

$$
\begin{array}{lrr}
20 \times 6 \times 15 \times 2 & = 3{,}600 & \\
12 \times 4 \times 6 \times 2 & = 576 & \\
2 \times 8 \times 12 \times 2\frac{1}{2} & = 480 & \\
\hline
 & 4{,}656 & 4{,}656 \\
\hline
 & & 26{,}034 \times 160 = 4{,}165{,}440 \text{ pounds.}
\end{array}
$$

Weight of woodwork, weight per cubic foot taken as 40 pounds:

$$
\begin{array}{ll}
\frac{2 \times 8}{144} \times 56.2 \times 75 \times 2 \times 40 & = 37{,}600 \text{ pounds of rafters.} \\
56.2 \times 104 \times 2 \times \frac{1}{12} \times 40 & = 39{,}000 \text{ pounds of roof boards.} \\
80 \times 100 \times \frac{2}{12} \times 40 & = 53{,}500 \text{ pounds of floor boards.} \\
\frac{2 \times 12}{144} \times 80 \times 75 \times 40 & = 40{,}000 \text{ pounds of roof joists.} \\
\text{Total weight of woodwork} & = 170{,}100 \text{ pounds.}
\end{array}
$$

Slate, weight per cubic foot taken as 170 pounds:

$$56.5 \times 104 \times 2 \times \frac{1}{48} \times 170 = 41{,}600 \text{ pounds.}$$

Plaster, weight per cubic foot taken as 90 pounds:

$$(360 \times 30 + 80 \times 40 + 100 \times 56.2 \times 2)\frac{3}{4} \times \frac{1}{12} \times 90 = 142{,}400 \text{ pounds.}$$

Air, weight per cubic foot taken as 0.08 pounds:

$$(80 \times 30 \times 100 + 80 \times 40 \times 100)0.08 = 32{,}000 \text{ pounds.}$$

Heat required:

$4{,}165{,}440 \times 50 \times 0.2100 =$	43,737,000 B.t.u.
$170{,}100 \times 50 \times 0.5700 =$	4,850,000 B.t.u.
$41{,}600 \times 50 \times 0.2159 =$	448,000 B.t.u.
$142{,}400 \times 50 \times 0.2000 =$	1,424,000 B.t.u.
$32{,}000 \times 50 \times 0.2415 =$	386,000 B.t.u.
	50,845,000 B.t.u.
Adding 15 per cent. for furnishings	7,627,000 B.t.u.
Total to raise to 50°	58,572,000 B.t.u.

The heating of the building structure may be very important in determining the size of the heating plant when a building is intermittently heated.

7. First Law of Thermodynamics.—When mechanical energy is produced from heat a definite quantity of heat is used up for every unit of work done and, conversely, when heat is produced by the expenditure of mechanical energy the same definite quantity of heat is produced for every unit of work spent. This *first law of thermodynamics* might also be called the law of the Conservation of Energy. The relation between work and heat has recently been determined with great accuracy and the results show that **one British thermal unit is equivalent to 778 foot-pounds.** This factor is called the **mechanical equivalent of heat** or Joule's equivalent.

Problems

1. Convert 50°F. to degrees Centigrade. Convert 150°C. to degrees Fahrenheit. Convert 219°F. to degrees Centigrade. Convert 225°F. to absolute temperature on the Fahrenheit scale.

2. A copper ball weighing 10 pounds is heated in a fire and immediately placed in a vessel of water having an equivalent water weight of 10 pounds. The water was raised in temperature from 50° to 100°. What was the temperature of the ball when it was removed from the fire?

CHAPTER II

HEAT LOSSES FROM BUILDINGS

8. Sources of Heat Loss.—When the interior of any building is maintained at a temperature higher than that of the outside air there is a continual loss of heat from the building. The functions of a heating system are, first, to **raise the temperature** of the interior of the building to the point desired and, second, to **maintain** this temperature by supplying sufficient heat to replace that lost from the building. The determination of the amount of heat lost from the building under maximum conditions is the first step in designing the heating system.

Before taking up the methods of calculating heat loss it is necessary to consider first the manner in which heat may be given up by any body. There are three ways in which heat can be transmitted from a body: by **radiation,** by **conduction,** and by **convection.** Each of these will be discussed separately.

9. Radiation.—Heat is transmitted, or **radiated,** through space by what is supposed to be a motion or vibration of the ether which is believed to pervade all space. Radiant heat follows the same physical laws as radiant light, being radiated, like light, in straight lines. We may have heat "shadows" just as we have light shadows and **the intensity of radiant heat is inversely proportional to the square of the distance from the source.**

Some substances are transparent to heat rays and others absorb them. Gases are almost perfectly transparent to radiant heat while such substances as wood, hair felt, and mineral wool are almost perfectly opaque to it. Radiant heat does not affect the medium through which it passes. When heat is radiated through the atmosphere, for example, the atmosphere is not perceptibly warmed by it.

The rate at which heat is radiated increases as the absolute temperature of its source is raised. It has been determined experimentally that the amount of heat radiated from a body varies as the 4th power of the absolute temperature, or

$$Q_r = KT^4$$

in which Q_r is the quantity of heat radiated, T the absolute temperature of the body, and K a constant depending upon the nature of the substance composing it. Radiant heat is given off by all bodies, the net amount of heat radiated by a body being the difference between the total amount radiated from it and the amount radiated from other bodies which is absorbed by it. If one body of absolute temperature T_1 is surrounded by another body of the same material at temperature T_2, then the heat which will pass between them is

$$Q_r = KT_1^4 - KT_2^4 = K(T_1^4 - T_2^4)$$

This is known as Stefan's law.

10. Conduction.—As has already been stated, heat will pass from any body to a body at a lower temperature which is brought into contact with it. It is further true that if one part of a body is at a higher temperature than another part there will be a flow of heat through the body. The transmission of heat in this manner is known as **conduction.** A familiar example of this phenomenon is the flow of heat along an iron bar, one end of which is heated in a fire. The ability of different materials to conduct heat differs considerably. Metals are the best conductors of heat, while such materials as wood, felt, asbestos, etc., are very poor conductors.

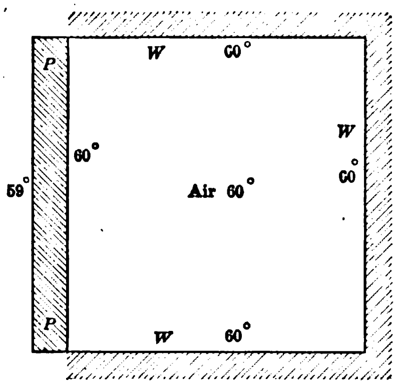


Fig. 2.

The conduction of heat which takes place through the walls of a building may be best understood from Fig. 2 in which PP is a plate, one side of which is enclosed by the walls WW. Let the temperature of the outside of the plate be 59° and let 60° be the temperature of the inside of the plate, of the inside walls WW, and of the inside air. Then all the heat that is lost by the room must be lost by conduction through the plate PP. The amount of heat lost will be dependent upon the material of the plate PP, upon the difference in temperature of its two sides, and upon its thickness.

Let E = the "specific conductivity" of the material.
t_1 = the temperature of the warmer side of the plate.
t_2 = the temperature of the cooler side of the plate.
A = the area of surface in square feet.
l = the thickness of plate in inches.
Q = the total quantity of heat transmitted.

Then

$$Q = \frac{AE(t_1 - t_2)}{l}$$

the **conductivity of the heat path** is then $\frac{AE}{l}$ and the **resistance** of the heat path is its reciprocal $\frac{l}{AE}$.

Example.—Suppose a boiler plate, 5 feet square, and ½ inch thick, to have a temperature of 70° on one side and 200° on the other side. Assume the specific conductivity of the metal to be 240 B.t.u. per hour per square foot of area per inch in thickness per degree difference in temperature. The total heat transmitted per hour is then

$$Q = \frac{25 \times 240(200 - 70)}{\frac{1}{2}} = 1{,}560{,}000 \text{ B.t.u. per hour.}$$

11. Convection.—When a body is in contact with a fluid at a lower temperature, the envelope of fluid surrounding it becomes heated by conduction of heat from the body. As this fluid envelope is heated its density decreases and it is forced to rise, giving place to the colder fluid from below. A continuous current is thus created and maintained over the surface of the body. This process of heat transfer is called **convection.** It should be noted that **the heat actually leaves the hot body by conduction** from its surface to the fluid in contact with it. The essential characteristic of the process of convection is the **continuous renewal of the fluid layer** at the surface of contact.

The loss of heat from a body by convection is independent of the nature of the surface of the body, and of the material composing it, but **is greatly affected by the form of the body,** a cylinder and a sphere, for example, transmitting different amounts of heat per square foot of surface. The velocity of the fluid over the surface also affects the rate of heat transmission. In the case of convection by air the air movement is often produced by some external force, as when the wind blows against a building or when a fan in an indirect heating system forces air over the surface of steam coils. An increase in the velocity produces a

more frequent renewal of the layer of air in contact with the body and augments the rate of heat transmission.

Heat may also be transmitted from a fluid to a solid by convection as well as from a solid to a fluid. An example of this process is the transfer of heat from the warm air of a room to the cold outside walls. The air, upon giving up its heat, increases in density and falls, giving place to warmer air from above and producing a continuous downward current.

12. Loss of Heat from Buildings.—The heat which is lost from a building may be divided into two parts: (*a*) the heat which passes by **conduction through the building structure**; and (*b*) the heat which is lost due to **air infiltration.** A third factor, the heat lost in warming air introduced for ventilation, might also be here mentioned.

The heat which flows by conduction through the walls and roof of the building is transmitted from the outer surface of the structure partly by radiation and partly by convection. The calculation of the heat lost by convection is very difficult. Methods of arriving at the loss by convection from bodies of various shapes were developed by Peclet and are given in Box's "Treatise on Heat," but these methods cannot, as a rule, be applied to the loss of heat from buildings. They assume, for example, that the air surrounding the object is, except for the influence of the heat from the body itself, in a perfectly quiescent state. In the case of buildings this is far from true, for the air surrounding a building is always circulated more or less rapidly by the winds. Because of the necessity of taking into account variable factors of this nature, the heat loss from a building could not be stated in any simple expression and the practical rules that are used for such calculations are therefore largely empirical. The common method of treating the conduction of heat through building walls as given in the following pages was translated by J. H. Kinealy from the work of Rietschel and published in the *Metal Worker.*

In the simplest form of building the walls consist of one solid piece of a single material and the transmission of heat takes place from the air of the room by convection, through the wall by conduction, and from the outer surface of the wall by convection and by radiation. Such a wall is shown in Fig. 3. In order that heat may flow through the wall it is necessary that the room temperature t_1 be higher than the temperature of the inside of the wall t_1',

that the temperature of the outside of the wall t_0' be lower than t_1'; and that the temperature of the outside air t_0 be lower than t_0'. The amount of heat which will be transferred from the air of the room to a unit area of the wall will be $a_1\ (t_1 - t_1')$ in which a_1 is a constant. The amount of heat flowing through a unit area of the wall will be $\frac{e_1}{x}\ (t_1' - t_0')$ in which e_1 is a constant which represents the specific conductivity of the material composing the wall. Similarly the heat transfer from a unit area of the outside wall surface is $a_0\ (t_0' - t_0)$.

When the rate of heat flow through the wall has reached a stable condition the quantity of heat flowing through successive

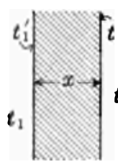


FIG. 3.

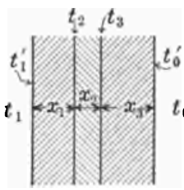


FIG. 4.

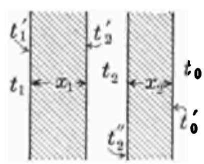


FIG. 5.

points of the wall thickness must be the same and we have, therefore,

$$a_1(t_1 - t_1') = \frac{e_1}{x}(t_1' - t_0') = a_0(t_0' - t_0)$$

A wall may be made up of a series of layers of different materials, as shown in Fig. 4. The transmission of heat takes place in the same way except that the conductivity of the successive layers may be different. In a wall such as shown in Fig. 5 the heat passes through the inside wall to the air in the air space and thence through the outside wall to the outside air, the temperature at each successive point from the inside to the outside being lower, as before. If a_1, a_2, a_3 and a_0 are the constants representing the conductivity of heat between the air and the wall surfaces (Fig. 5) and e_1 and e_2 are the specific conductivities of the materials composing the two walls, then the heat transmitted through the walls may be expressed in any of the following equal forms:

$$a_1(t_1 - t_1') = \frac{e_1}{x}\ (t_1' - t_2') = a_2(t_2' - t_2) = a_3(t_2 - t_2'')$$

$$= \frac{e_2}{x_2}\ (t_2'' - t_0') = a_0(t_0' - t_0)$$

In order to use these expressions it would be necessary to know the temperature of all the wall surfaces. These temperatures are not known. The only known temperatures are the temperatures of the air inside the room and of the air outside of the building. Therefore, let us assume that the heat transmission through the wall may be represented by the expression $k(t_1 - t_0)$, in which k is a constant to be determined. We then have for Fig. 3:

$$k(t_1 - t_0) = a_1(t_1 - t_1') = \frac{e_1}{x}(t_1' - t_0) = a_0(t_0' - t_0)$$

And for Fig. 5:

$$k(t_1 - t_0) = a_1(t_1 - t_1') = \frac{e_1}{x_1}(t_1' - t_2') = a_2(t_2' - t_2)$$

$$= a_3(t_2 - t_2'') = \frac{e_2}{x_2}(t_2'' - t_0') = a_0(t_0' - t_0)$$

Solving for k we have, for Fig. 3:

$$k = \frac{1}{\frac{1}{a_1} + \frac{x}{e_1} + \frac{1}{a_0}} \tag{1}$$

And for Fig. 5:

$$k = \frac{1}{\frac{1}{a_1} + \frac{x_1}{e_1} + \frac{1}{a_2} + \frac{1}{a_3} + \frac{x_2}{e_2} + \frac{1}{a_0}} \tag{2}$$

For thin glass or thin metal walls $\frac{x}{e}$ is a very small quantity and may be neglected.

The values of a and e must be known before k can be determined. The value of the convection factor, a, is determined by Grashof by the following equation:

$$a = c + d + \frac{(40c + 30d)T}{10{,}000}$$

in which c is a factor depending on the condition of the air, whether at rest or in motion. Rietschel gives the following values for c:

TABLE III.—VALUES OF c

	c
Air at rest, air in rooms	0.82
Air with slow motion, air in rooms in contact with windows	1.03
Air with quick motion, air outside of a building	1.23

The factor d depends upon the material composing the wall and on the condition of the surface. The values for d may be taken as follows:

TABLE III.—VALUES OF d

Substance	d	Substance	d
Brickwork	0.740	Sheet iron	0.570
Mortar and similar materials	0.740	Sheet iron polished	0.092
Wood	0.740	Brass polished	0.053
Glass	0.600	Copper	0.033
Cast iron	0.650	Tin	0.045
Paper	0.780	Zinc	0.049

T is the difference between the temperature of the air and that of the surface of the wall. For walls composed of materials of low conductivity or very thick walls it may be taken as zero. In approximate calculations it is usually taken as zero.

The following values of T are given by Rietschel:

TABLE IV.—VALUES OF T

Brickwork 5 inches thick	14.4
Brickwork 10 inches thick	12.6
Brickwork 15 inches thick	10.8
Brickwork 20 inches thick	9.0
Brickwork 25 inches thick	7.2
Brickwork 30 inches thick	5.4
Brickwork 40 inches thick	1.8
For single windows	36.0
For double windows	18.0
For wooden doors	1.8

Table V gives values of e. These values, as given by different authorities, vary considerably.

TABLE V.—VALUES OF e

	e
Brickwork	5.60
Mortar, plaster	5.60
Rubble masonry	14.00
Limestone	15.00
Marble, fine-grained	28.00
Marble, coarse-grained	22.00
Oak across the grain	1.71
Pine, with the grain	1.40
Pine, across the grain	0.76
Sandstone	10.00
Glass	6.60
Paper	0.27

For example, assume a brick wall as shown in Fig. 6. There are four air contact surfaces and two walls through which conduction takes place, then:

k is the same as in equation (2).

Rietschel assumes a_1, a_2, and a_3 equal and he uses the same value of T as for a solid of thickness equal to the brickwork without the air space.

$$a_1 = a_2 = a_3 = 0.82 + 0.74 + \frac{(40 \times 0.82 + 30 \times 0.74)10}{10{,}000} = 1.62$$

$$a_0 = 1.23 + 0.74 + \frac{(40 \times 1.23 + 30 \times 0.74)10}{10{,}000} = 2.04$$

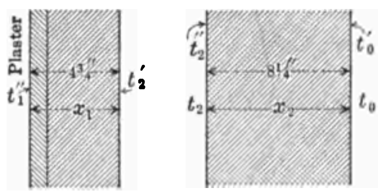


FIG. 6.

Since both walls are of brickwork

$$\frac{x_1}{e_1} = \frac{4.75}{5.6} = 0.85$$

$$\frac{x_2}{e_2} = \frac{8.25}{5.6} = 1.47$$

Substituting in equation (2)

$$k = \frac{1}{0.62 + 0.85 + 0.62 + 0.62 + 1.47 + 0.49} = 0.214$$

Making this same calculation, assuming $T = 0$, gives

$$k = 0.210$$

In Table VI are given the values of k for various building materials which have been determined either experimentally or by methods similar to the foregoing, by different authorities. A more complete table of values of k is given in the Appendix.

Table VI.—Coefficients of Heat Transmission for Various Materials

	k B.t.u. per square foot, per hour per degree difference in temperature
Walls:	
Brick wall 4 inches thick, plain	0.52
Brick wall 8½ inches thick, plain	0.37
Brick wall 4 inches thick, furred and plastered	0.28
Brick wall 8½ inches thick, furred and plastered	0.23
Concrete wall 4 inches thick, furred and plastered	0.31
Concrete wall 6 inches thick, furred and plastered	0.30
Clapboard wall with paper, sheathing, studding, and lath and plaster	0.23
Ceilings and Roofs:	
Lath and plaster, no floor above	0.32
Lath and plaster, single floor above	0.26
Tin or copper roof on 1-inch boards	0.45
Shingle roof	0.33
Windows, Skylights and Doors:	
Ordinary windows	1.09
Double windows	0.45
Single skylight	1.50
Pine door ¾ inch thick	0.47
Oak door ¾ inch thick	0.63

13. Temperatures Assumed in Heating.—In determining the heat transmission through the walls of a building, it is necessary to assume certain temperatures for the outside air and for the inside air. In the latitude of New York City it is customary to assume 0° for the outside temperature. In the latitude of Washington it is customary to assume 20° above, and in the latitude of St. Paul 20° below. The assumed outside temperature is ordinarily taken as the temperature which might exist for a period of at least 24 hours. The inside temperature to be assumed depends upon the type of building. The temperature maintained in many classes of buildings is largely a matter of custom. In residences this temperature is higher in the United States than in any other country in the world, with the possible exception of Germany. In England and many other countries a temperature of from 55° to 60° is a perfectly proper temperature for a room; while in this country the temperature ordinarily ranges from 65° to 70°.

The following are the inside temperatures usually assumed:

2

TABLE VII.—INSIDE TEMPERATURES

	Degrees
Residences	70
Lecture rooms and auditoriums	65
Factories for light work	65
Factories for heavy work	60
Offices and schools	68 to 70
Stores	65
Prisons	65
Bathrooms	72
Gymnasiums	55 to 60
Hot houses	78
Steam baths	110
Warm air baths	120

The following assumptions are ordinarily made for unheated spaces:

TABLE VIII

	Degrees
Cellars and closed rooms	32
Vestibules frequently opened to the outside	32
Attics under a roof with sheathing paper and metal or slate covering	25
Attics under a roof with paper sheathing and tile covering	32
Attics under a roof with composition covering	40

14. Heat Lost Due to Infiltration.—No building is ever air-tight; there is a large amount of leakage through the walls, the windows, and other openings. The amount of this infiltration depends largely upon how well the building has been constructed and upon the type of construction. For this reason no definite rule can be given for the determination of infiltration, and the allowance made for this loss must be a matter of judgment and experience. Usually the volume of infiltration is expressed as a certain ratio of the cubic contents, and experiments go to show that the air of the average room is changed about once an hour because of infiltration. In rooms where doors are frequently opened to the outside, or where the windows are loosely fitted and the construction is faulty, the change of air may be as frequent as twice an hour.

Strictly speaking, however, the amount of infiltration does not depend upon the volume of the room but upon the nature and size of the windows. Experiments[1] have shown that the amount

[1] See "Window Leakage" by S. F. VOORHEES and H. C. MEYER, *Trans.* A. S. H. & V. E., 1916.

of air leakage varies considerably for different types of windows. Some forms of metal sash allow a large amount of leakage to take place. Weather strips are very effective in reducing air leakage. As the principal source of leakage is around the window sash the amount of leakage may be considered as varying directly with the **perimeter** of the windows. It is customary to assume a leakage of from 1.0 to 1.5 cubic feet of air per minute per foot of sash perimeter for windows equipped with weather strips. For windows without weather strips a considerably higher factor should be used. In large buildings the amount of infiltration should be computed in this manner, especially in the case of a tall or exposed building.

The heat required to supply these infiltration losses must be sufficient to warm the air from the temperature of the outside air to that of the room. If the infiltration is figured on the basis of a certain number of air changes per hour the loss from this source may be expressed as follows:

Let H_a = heat required per hour to supply loss due to infiltration.
C = cubic contents of the room.
n = number of changes per hour.
t_r = temperature of the room.
t_0 = temperature of the outside air.

$$H_a = \frac{C(t_r - t_0)n}{55.2}$$

The factor $55.2 = \frac{1}{0.2415 \times 0.0749}$ = heat required to raise the temperature of 1 cubic foot of air 1° where 0.2415 is the specific heat of air at constant pressure and 0.0749 is the weight of a cubic foot of air at 70°.

15. Heat Required for Ventilation.—The heat required for ventilation can easily be computed when the amount of air supplied per hour is known.

Let H = heat required for ventilation.
Q = quantity of air supplied in cubic feet per minute.

Then,

$$H = \frac{60 \times Q(t_r - t_0)}{55.2}$$

Besides supplying heat to replace that lost through the walls and by infiltration of air, a heating system must supply the heat

which is stored in the structure and its contents and in the inside air. In heavy buildings the effect of the heat stored in the walls may have a material effect upon the amount of heat which must be supplied to warm the building initially. If the building is intermittently heated the effect is decidedly appreciable. The best illustration is in the cathedrals of Europe in which no heating systems are used and the heat stored in the walls during the summer serves to keep the building warm throughout the year.

The heat which is required initially to warm the inside air and the building structure affects the rapidity with which the building can be heated to the desired temperature. It is often desirable to investigate this question in designing a heating system which is to be operated intermittently and to increase the radiation, if necessary, so that the building can be warmed within a reasonable time.

16. Calculation of Heat Loss from a Building.—In determining the heat loss from a room all surfaces should be considered which have on the outside a lower temperature than the temperature to be maintained in the room. If the room is over a portion of the basement which is unheated or below an unheated attic, the loss through the floor or ceiling should be considered. Similarly, if an adjacent room is liable to be unheated at times, the additional heat loss through the wall should be taken into account. Ordinarily it is assumed that there is no loss through inside walls where the surrounding rooms are heated.

The conditions under which the room is to be used should be studied in determining the amount of heat necessary. In certain rooms such as restaurants in the basements of buildings, for example, where there are no outside windows, the problem is often one of cooling rather than heating. In designing any heating system, careful consideration should be given to the conditions existing, and to the exposure of each room in the building.

The first step in computing the heat loss is to determine for every room the gross surface of exposed wall, and the window surface, from which the net wall surface is obtained by subtraction. The heat loss through the walls can then be computed from the expression,

$$H_w = Wk(t_r - t_0)$$

in which

H_w = heat loss in B.t.u. per hour.
W = exposed wall surface in square feet.
t_r = inside temperature.
t_0 = outside temperature.
k = coefficient of heat transmission.

A similar expression must be worked out for the walls, ceilings and floors next to unheated spaces. The value of t_r in such cases may be taken from Table VII.

The heat loss through the glass surface is computed from the expression,

$$H_g = Gk(t_r - t_0)$$

in which G is the area of the glass surface in square feet and k is the heat transmission for glass.

The heat lost due to air infiltration is next determined by one of the methods given on pages 18 and 19.

The total heat loss from the room in B.t.u. per hour is then

$$H = H_w + H_g + H_a$$

17. Correction Factors.—The heat losses determined by this method are for rooms not exposed to prevailing winter winds.

For exposed rooms it is customary to add certain percentages to the heat losses to allow for extreme exposures. Also, when a building is intermittently heated, an allowance should be made to insure that the building can be heated within a reasonable time. The correction factors commonly used are given in Table VIII.

TABLE VIII.—FACTORS FOR EXPOSURE AND INTERMITTENT HEATING

	Percentage to be added
For exposure in direction of prevailing winter winds (usually north and northwest)	15
Same, severe conditions	20
For west exposure	10
For building heated during the day only and closed at night	15
For buildings heated during the day and open at night	30
For buildings heated intermittently	50

18. Approximate Rules for Determining the Loss of Heat.—A common rule for the loss of heat from a building is that given by Prof. R. C. Carpenter in his book on "Heating and Ventilation." This rule is developed from the following consideration: Referring to Table VI we notice that 1 square foot of glass conducts

approximately four times as much heat as a plastered brick wall 4 inches thick. If, then, we divide the wall surface by 4, the result will give us the number of square feet of glass surface, which would lose the same quantity of heat. Adding to this the actual glass surface would give us the total equivalent glass surface. As the heat loss per square foot of glass surface per degree difference in temperature is approximately 1 B.t.u. per hour, this total equivalent glass surface multiplied by the temperature difference gives the heat lost through the walls. In considering the infiltration losses it is assumed that for ordinary-sized rooms the air in the room will be changed once an hour. One cubic foot of air weighs, approximately, $\frac{1}{13}$ pound. To raise a pound of air 1° would require about 0.0183 B.t.u. or one heat unit will heat in round numbers about 55 cubic feet of air 1°. If, then, we divide the contents of a room by 55 we will have the heat lost by filtration through the walls per degree difference in temperature. Adding these factors together will give the total heat lost from the room. This rule may be concisely expressed as follows:

Let H = B.t.u. loss per hour.
G = glass surface in square feet.
W = net exposed wall surface in square feet.
C = cubic contents of room.
n = number of times the air in the room is changed per hour.

$$H = \left(\frac{Cn}{55} + \frac{W}{4} + G\right)(t_r - t_0)$$

The quantity n ordinarily varies from 1 to 3; for ordinary rooms $n = 1$; for corridors $1\frac{1}{2}$; for vestibules 2 to 3.

This rule will indicate an excessive heat loss where a room has large cubic contents and small window surface and will show heat losses that are too small where the room has a very large amount of exposed surface in proportion to its cubic contents. As the infiltration loss in a room depends upon the outside wall and window surface, the following rule seems somewhat more rational.

Using the same notation as before,

$$H = \left(\frac{W}{4} + G\right)(t_r - t_0)n$$

where n is the infiltration factor.

The factor n has been determined by comparison with many successful plants that have been installed and it has been found to vary from $1\frac{1}{2}$ to $2\frac{1}{2}$. For ordinary rooms $n = 1\frac{1}{2}$; for

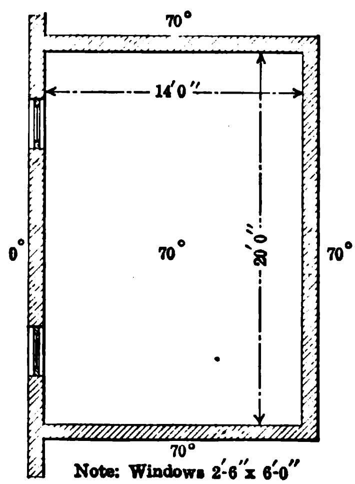
corridors $n = 2$; for vestibules and rooms where doors are opened frequently $n = 2$ to $2\frac{1}{2}$.

19. Heat Given Out by Persons and Processes.—In considering the amount of heat necessary to heat a room attention must be given to the amount of heat that will be given off by the occupants of the room or by the processes which go on in it. But these sources of heat cannot always be depended upon, as it may sometimes be necessary to heat a room when there are no people in it or when the processes ordinarily going on are not in operation. On the other hand, it may be necessary to cool the room instead of heat it. Often in large auditoriums the greatest source of heat in a room are the people in it. The following table shows the heat given off by the human body under various conditions in a room at a temperature of 70°.

Table IX

	B.t.u. per hour
Adults at rest	380
Adults at work	450
Adults at violent exercise	600
Children	240
Infants	63

Example 1.—Assume a room, as shown in Fig. 7. Let the temperature be maintained in the room at 70°, the temperature of the outside air be 0°. Let the walls be of brick, 18 inches thick, plastered on the inside, the windows be $2\frac{1}{2}$ by 6 feet, the ceiling of the room be 10 feet high. Let the room be on the second floor of the building, the rooms above and below heated. The window surfaces are $2 \times 2\frac{1}{2} \times 6 = 30$ square feet. The gross wall surface is $20 \times 10 = 200$ square feet. The net wall surface is $200 - 30 = 170$ square feet. The cubic contents is $20 \times 14 \times 10 = 2800$ square feet. Then the heat lost from the room would be determined as follows.



Note: Windows 2'-6" x 6'-0"

Fig. 7.

By the B.t.u. method:

$$H_w = 170 \times 0.24\ (70 - 0) = 2856$$
$$H_g = 30 \times 1.09\ (70 - 0) = 2289$$
$$H_a = \frac{2800\ (70 - 0)}{55.2.} \times 1.0 = 3551$$
$$H = 8696 \text{ B.t.u. per hour.}$$

By Carpenter's rule:

$$\begin{aligned} H &= \left(\frac{2800 \times 1}{55} + \frac{170}{4} + 30\right)(70 - 0) \\ &= (50.9 + 42.5 + 30) \times 70 \\ &= \qquad 8638 \text{ B.t.u. per hour.} \end{aligned}$$

By Allen's rule:

$$\begin{aligned} H &= \left(\frac{170}{4} + 30\right)(70 - 0)\ 1.5 \\ &= (42.5 + 30) \times 70 \times 1.5 \\ &= \qquad 7613 \text{ B.t.u. per hour.} \end{aligned}$$

Problems

1. Compute the value of k for a wall consisting of 2 inch pine boards. Assume $T = 3$.

2. Compute the heat loss per hour, per square foot of area, of a wall consisting of two thicknesses of 1 inch pine boards with an air space of 2 inches between. Room temperature 60°, outside temperature 10°. Assume $T = 1.8$.

3. Compute the heat loss per hour, per square foot of area, of a wall consisting of 1 inch oak boards, an air space of 1 inch, and 4 inches of brickwork.

4. In the room of Fig. 7 (Example 1) find the percentage of the heat loss which would be saved during a heating season of 8 months if double windows were used. Assume average temperature of the room and the surrounding rooms to be 65° and the average outside temperature to be 40°.

5. Taking the same room as in Example 1, heated to a temperature of 60°, with the surrounding rooms at 70° and the air outside at 10°, how much heat must be supplied to the room per hour? Inside walls are of lath and plaster. Ceiling is of lath and plaster, with single floor above, and the room below has its ceiling plastered.

6. Take the same room as Example 1, except that the room is covered by a flat tin roof. The air space between the ceiling of the room and roof should be assumed to be at a temperature of 32°.

CHAPTER III

DIFFERENT METHODS OF HEATING

20. Classification of Heating Systems.—The different types of heating systems may be classed under two general heads: **direct** and **indirect.** In direct heating the heating surfaces are placed in the rooms to be heated. Under this head come grates, stoves, steam radiators, and hot-water radiators. In indirect heating systems the heating surfaces are placed outside the rooms to be heated and air passes over them, is heated, and flows to the various rooms through pipes or flues. Hot-air furnaces would be included under this head, together with various systems of heating in which fresh cold air is made to pass over steam or hot-water radiators on its way to the rooms.

Indirect systems may be subdivided into two classes: those in which the air circulates by gravity and those in which the circulation is produced by a fan or some other mechanical device. A good example of the gravity or "natural" systems is the hot-air furnace in which the circulation of air through the furnace and air ducts is produced by the difference in temperature, and consequently in density, between the air in the hot-air ducts and the cold air outside. The fan systems of heating used in schools and churches are examples of the forced-circulation type in which the circulation is produced by a disc fan or a pressure blower. Before studying the design of the various systems of heating it is desirable to understand in general their advantages and disadvantages.

21. Grates.—The most primitive form of heating apparatus is the grate. In the grate the air which passes through the fire, and is heated by the fire, all passes up the chimney and only the heat given off by radiation to the walls and objects in the room and the small amount given off by the chimney walls is effective in heating the room. In grates of better construction this condition is somewhat improved by surrounding the grate with firebrick so arranged that it becomes highly heated and radiates heat to the room. But the fact that all the air heated by the grate passes up

the chimney makes the grate a very uneconomical form of heating. In the best forms of open grates only about 20 per cent. of the heat of the fuel is effective in heating the room. This form of heating, however, is highly recommended by many and is a very popular method of heating throughout England and Scotland. The feeling of a grate-heated room is quite different from that of a room heated by other means. All of the heat is given off by radiation and the air is at a considerably lower temperature than the objects in the room, owing to the fact that the radiated heat does not heat the air through which it passes. The air of the room being at a much lower temperature, its capacity for moisture is not increased as much as it would be were the air heated to a higher temperature. The result is that the air contains proportionately more moisture than is the case with most other forms of heating, which, no doubt, is an advantage. On the other hand, it is impossible to heat the room uniformly and a person is either hot or cold, depending on his distance from the fire. Heating by means of grates is practised only in the more moderate climates. Grates are useful in houses heated by other means, as the open chimney forms a most efficient foul-air flue and greatly improves the ventilation.

22. Stoves.—The stove is a marked improvement over the grate, particularly from the standpoint of economy. The modern base-burner stove is one of the most efficient forms of heating apparatus, making use of from 70 to 80 per cent. of the heat in the fuel. In heating a room, the hot surface of the stove, being at a higher temperature than that of the surrounding objects in the room, radiates heat directly to those objects. In addition, heat is given to the air of the room by contact with the hot surface of the stove. In selecting a stove to heat a given room care should be taken to choose one of ample size so that only in the coldest weather would it be necessary to keep the drafts wide open in order to heat the room. At the present time the stove as a general source of heat is being rapidly discarded because of the attendance required, the space occupied, the unsightly appearance of the stove, and the fact that a separate stove is required in every room for satisfactory results. Another objection to the stove is the fact that it does not provide ventilation to the room which it heats.

23. Hot-air Furnaces.—The hot-air furnace is the natural outgrowth of the stove. In this system one large furnace is placed in the basement of the building, and the air is taken

from the outside or recirculated from the house, passed over the surfaces of the furnace, and carried up through the flues to the rooms to be heated. The principle advantages of the hot-air furnace are that it provides a cheap method of furnishing both heat and ventilation, requires little attendance, and does not deteriorate rapidly when properly taken care of. The greatest disadvantage of this system is that the circulation of the heated air depends entirely upon natural draft; that is, it depends upon the difference in weight between the air inside the flues and the air outside the flues. This difference is extremely small, so that the force producing circulation in the flue is always small. When a very strong wind blows against one side of the house, air from the outside enters through the window cracks and other small openings, forming a slight pressure in the rooms and preventing the warm air from entering, thus making it difficult to heat the rooms on that side of the house. If the system is carefully designed, however, this difficulty can be overcome in a measure. Another serious objection to the hot-air furnace is that it is seldom dust-tight, and dust, ashes, and gases from the fire are carried into the rooms. In general, the hot-air furnace may be considered as a very good type of heating plant for small residences, but because of the small force available for producing circulation its use is limited to buildings where the length of the horizontal flues does not exceed 15 feet.

In the case of the hot-air furnace, the heat is carried from the furnace by the air which passes around the furnace and then enters the rooms through the flues. This air circulates in the room and heats the contents of the room and supplies the heat which is lost through the walls. The economy of the hot-air system will vary, depending on the relative proportions of the air taken from the outside and from the rooms. If the air entering the furnace is taken from the house and not from the outside, the economy of the hot-air furnace will be about the same as that of the steam system. If, however, cold air be taken from the outside, an additional amount of heat will be used in heating this cold air up to the temperature of the rooms. Control of the heat supply, with a hot-air furnace, is readily obtained by adjusting the dampers at the registers in each room and by manipulating the furnace drafts.

24. Direct Steam Heating.—From the standpoint of ventilation, direct steam heating, without other means for ventilation,

is not as desirable as the hot-air furnace. Mechanically, however, it has many advantages. The modern radiator is easily adapted to almost any location in the room and its operation is not affected by the winds. The circulation of the system is positive and a distant room can be heated as easily as those close to the boiler.

In the older forms of direct steam-heating systems control of the heat supply is difficult because the radiators, being large enough to heat the room on the coldest days, give off too much heat for average conditions. Since the entire radiating surface is heated to a high temperature when the radiator is turned on, much manipulation of the valves is required in order to keep the room at a comfortable temperature. In recent years these disadvantages have been overcome in the so-called "vapor" systems which make use of steam at pressures but slightly higher than atmosphere, and in some cases below atmosphere. In these systems the steam supply to each radiator can be controlled at the inlet valve so that only the quantity actually required is admitted to the radiator, and much better regulation is therefore possible. The efficiency of the direct steam-heating system in a well-designed plant is from 60 to 70 per cent.

25. Direct Heating by Hot Water.—The application of direct hot-water radiators as a method of heating is similar to that of steam, with the exception that the surfaces are usually at a much lower temperature and more radiating surface is therefore required. Hot-water systems are preferable to ordinary steam systems in that the temperature of the radiating surfaces can be easily controlled, and can be anywhere from the temperature of the room to 190°, or even higher in the case of certain forms of hot-water systems. Another advantage is that the surface of the radiator, being at a lower temperature, gives off more heat by convection and less by radiation, which tends to keep the room at a more uniform temperature throughout and makes it more comfortable to the occupants. The principal disadvantage of the hot-water system lies in the fact that the circulation of the system is ordinarily produced only by the difference in weight between the water in the hot leg of the system and that in the cold leg of the system. The difference in temperature between the two legs is small, being usually about 10° to 20°, so that the resulting force producing circulation is there-

fore small. It is necessary to be very careful in designing the piping for a hot-water system as the circulation may be easily affected by the friction in the piping and the height of the radiator above the boiler. The greater the height above the boiler the greater will be the difference in weight between the two columns of water and the stronger will be the force producing circulation. This system in general requires more careful design and construction than the steam system. Another disadvantage is that, because of the great thermal capacity of the water contained in the system, considerable time is necessary to change its temperature and the system cannot be made to respond quickly to sudden changes in the demand for heat. The efficiency of the hot-water system is practically the same as that of a steam system and we may expect to obtain in the rooms about 60 or 70 per cent. of the heat in the fuel.

Where hot-water heating is used in large buildings the circulation is produced by a pump. The difficulty of circulation is then done away with and the flow of water is certain and rapid.

26. Indirect Steam and Hot-water Heating by Natural Circulation.—In heating with indirect steam or water radiation cold air is drawn from the outside, passed through and around the hot radiator, which is usually situated in the basement, and delivered through flues to the rooms to be heated. The rules governing the introduction of air into the rooms and the method of running the pipes are similar to those employed in the installation of the hot-air furnace. The principal advantages of indirect steam and water heating over the hot-air furnace are that each room has a separate source of heat, the system is not affected by the winds, and no dust or obnoxious gases are carried to the rooms. The source of heat being independent of the position of the boiler, it is possible to place the indirect radiators anywhere in the building and long air flues are not necessary. This makes the indirect radiator much more certain in operation than the hot-air furnace. The application of indirect hot-water radiators is similar to that of steam radiators and the economy is practically the same, although the use of hot water for indirect heating has been much more limited than the use of steam. The installation of hot-water radiators must be done with great care so that each radiator will at all times have the proper amount of water circulating through it, for if for any reason the circulation is stopped the water in the radiator will be

in danger of freezing. In mild climates this difficulty would not be as serious as in locations where the weather is extremely cold.

27. Fan Systems of Heating.—In buildings of a public or semi-public character, where a large number of people are gathered in a relatively small space, it is necessary to provide adequate ventilation. With the systems that have been previously described it is impossible to introduce sufficient quantities of air to ventilate such buildings properly. It may be said in general that no system of natural circulation has ever produced satisfactory ventilation in a room occupied by a large number of people; it is necessary to provide some mechanical means for introducing the air. In fan systems the pressure produced by the fan makes the circulation positive so that it is not affected by winds or by the distance of the room from the source of heat. The air is taken from the outside, or sometimes recirculated from the inside, and is passed through the heating coils and forced into the building by the fan.

There are three general methods of heating and ventilating with the fan system. In one system the air is first passed through a tempering coil and then taken by the fan and delivered through a heating coil. Each room has a connection both to the hot air and to the tempered air chambers. The temperature of the air in the room is adjusted by taking the air partly from the hot-air chamber and partly from the tempered-air chamber. In the second system the rooms themselves are heated by means of direct radiation and the fan delivers air to the rooms only for the purpose of ventilation. In this case a much smaller amount of heating surface in the fan system is necessary as the air is heated to only about 70°. The economy of this system is also better, due to the fact that it is necessary to run the fan only when it is necessary to ventilate the building.

In the third system both the heating and ventilating is done by means of the fan system but only one system of ducts is installed. The temperature of the air leaving the heating coils is adjusted so as to maintain the proper room temperature. This method is applicable only in factory buildings, theatres, and other buildings which are divided into only a few rooms, making it possible to utilize air of the same temperature throughout the entire building.

28. Combinations of Different Systems.—In addition to the combination just described, of direct radiation and fan ventila-

tion, there have been devised innumerable combinations—combinations of direct and indirect steam systems, direct and indirect hot water, water and hot air, and steam and hot air. The combinations which have been most used are those of direct and indirect steam systems and of hot water and hot air.

29. Economy of Heating Systems.—The economy of any heating system depends upon the completeness with which the heat in the fuel is effectively utilized in heating the building. The principal sources of loss and the manner in which the heat is utilized in any type of heating system are as follows:

Losses:

Imperfect combustion.
Sensible heat in the chimney gases.
Combustible in the ash.
Radiation from boiler or furnace.
Radiation from flues or piping.
Losses through excessive temperature in the building.

Heat utilized:

Heat utilized in supplying the heat losses from the building.
Heat used for ventilation.

Of the losses, the first three are dependent rather upon the design of the grates and firepot than upon the type of heating system. The radiation from the boiler or furnace is partially recovered as it serves to warm the basement and decreases the heat loss to the basement from the rooms above. The loss from this source is fairly constant, regardless of the amount of heat delivered by the boiler or furnace and if a very low fire is carried, as in mild weather, it may become quite appreciable in comparison with the heat delivered. The loss from the flues or piping is also partially utilized in warming the building.

The heat used to supply the heat losses from the building is the principal product of any heating system. A part of this heat may be considered as a loss, however, if excessive temperatures are maintained either during the hours when the building is occupied, or during the night or other times when a low temperature could be carried.

The amount of heat used for ventilation will depend upon the amount of fresh air supplied. The air introduced for ventilation is discharged from the building at room temperature, and the heat contained in this air in excess of the heat in the outside air

is evidently the amount chargeable to ventilation. While this item might, from the standpoint of heating only, be considered as a loss, it is really the price that must be paid for good ventilation which is essential to health and comfort. In many States there are laws which specify the minimum amount of air which must be furnished per hour for each occupant in theatres and other buildings of a public character. The necessity and importance of ventilation will be discussed in later chapters.

CHAPTER IV

PROPERTIES OF STEAM

30. The Formation of Steam.—The different types of heating systems discussed in the previous chapter owe most of their characteristic features to the element used to transmit the heat from the boiler or furnace to the rooms. The most important is the steam system in which steam serves as the medium for carrying the heat from the boiler to the radiators. Before taking up the design of steam-heating systems it is necessary to study the nature and properties of steam.

Steam as produced in the ordinary boiler contains a certain amount of water in suspension as does the atmosphere in foggy weather. Let us suppose that we have a boiler partly filled with cold water, and that heat is applied to the outside of the boiler. As the water in the boiler is heated its temperature slowly rises until a certain temperature is reached at which small particles of water are changed into steam. The steam bubbles rise through the mass of water and escape from the surface. The water is then said to **boil.** The temperature at which the water boils **depends entirely upon the pressure in the boiler.** The steam produced from the boiling water is at the **same temperature** as the water, and under this condition the steam is said to be **saturated.** If we close the steam outlet the pressure in the boiler and the temperature of the water and steam will increase rapidly. If we continue to apply heat to the boiler with the outlet partly closed so that a constant pressure is maintained, the temperature of the steam and water will remain constant until all of the water is evaporated into steam. Any further addition of heat will raise the temperature of the steam above the boiling point and it will then be **superheated.**

31. Superheated Steam.—**Superheated steam is steam at a temperature higher than the temperature of the boiling point corresponding to the pressure.** If water were to be intimately mixed with superheated steam some of the heat in the steam would be used in evaporating the water and the temperature of

the steam would be lowered. If sufficient water were added the superheat would be entirely used up in evaporating the water and the steam would then be saturated. Superheated steam can have any temperature higher than that of the boiling point. When raised to any temperature considerably above the boiling point it follows very closely the laws of a perfect gas and may be treated as a perfect gas.

32. Saturated Steam.—When steam is at the temperature of the boiling point corresponding to its pressure it is said to be saturated. If this saturated steam contains no suspended moisture it is said to be dry saturated steam, or in other words, **dry saturated steam is steam at the temperature of the boiling point and containing no water in suspension. If heat is added** to dry saturated steam, not in the presence of water, it will become **superheated. If heat is taken away** from dry saturated steam it will become **wet steam.** The steam used in a heating plant is saturated steam and nearly always contains moisture, so that the substance used as a heating medium is really a **mixture of steam and water.** Steam at a pressure equal to or slightly above atmosphere is commonly known as **vapor.** It should be remembered, however, that **the difference between vapor and steam is merely one of pressure,** and that vapor is in no sense a separate state of the substance. **Dry saturated steam is not a perfect gas** and the relations of its pressure, volume, and temperature do not follow any simple law but have been determined by experiment. The properties of dry saturated steam were originally determined by Regnault between 60 and 70 years ago, and so carefully was his work done that no errors in his results were apparent until within very recent years, when the great difficulty of obtaining steam which is exactly dry and saturated became appreciated, and new experiments by various scientists proved that Regnault's results were slightly high at some pressures and slightly low at others.

33. Properties of Steam.—The heat used in the formation of 1 pound of superheated steam at any pressure from water at 32° may be divided into three parts: (*a*) the **heat of the liquid,** which is the heat required to raise the temperature of the water from 32° to the temperature of the boiling point; (*b*) the **latent heat of vaporization,** which is the amount required to change the 1 pound of water at the temperature of the boiling point to dry saturated steam at the same temperature; and (*c*) the "heat of superheat" or, more simply, the **superheat,** which is the heat added

to 1 pound of steam to raise it from the boiling point temperature to the final temperature.

34. Heat of the Liquid.—The heat of the liquid may be determined for any boiling point temperature by the expression

$$h = c(t - 32)$$

in which

h = the heat of the liquid.
t = the boiling point temperature.
c = the specific heat of water.

For approximate results c may be taken as $= 1$. The change in the volume of the water during the increase in temperature is extremely small, and the amount of external work done may be neglected and all of the heat of the liquid may be considered as going to increase the heat energy of the water.

The heat of the liquid, together with the other properties of saturated steam, is given in Table X for various steam pressures. This table is condensed from Marks and Davis' complete tables which are generally accepted as being accurate.

35. Latent Heat.—The latent heat of steam has been defined as the **heat required to convert 1 pound of water at the temperature of the boiling point into dry saturated steam at the same temperature.** Experiments show that the latent heat, usually designated by L, diminishes as the pressure increases.

When water is changed into steam, the volume is greatly increased, so that a considerable portion of the latent heat is used in doing external work. The remainder may be considered as being utilized in changing the physical state of the water. Let P be the pressure at which the steam is generated, V the volume of 1 pound of steam, and v the volume of 1 pound of water; then the external work done is equal to

$$P(V - v)$$

At 212° the external work done in producing 1 pound of steam is equivalent to 73 B.t.u. or about one-thirteenth of the latent heat.

Experiments show that the latent heat of steam diminishes about 0.695 heat units for each degree that the temperature of the boiling point is increased. If t be the temperature of the boiling point, then, approximately,

$$L = 1072.6 - 0.695(t - 32)$$

When steam condenses the same amount of heat is given up as was required to produce it.

36. Total Heat of Steam.—The **total heat of dry saturated. steam** is the heat required to change 1 pound of water at 32° into dry saturated steam. This quantity will be designated by H, and

$$H = h + L$$

The experimental results given in the table for the value of the total heat may be approximated very closely by means of the formula

$$H = 1072.6 + 0.305(t - 32)$$

It is more accurate, however, to take the values of the total heat from the tables than it is to compute them from the formula. The total heat in 1 pound of steam under any condition of moisture or superheat is the amount of heat required to change it from water at 32° to its existing condition.

When steam contains entrained water the percentage by weight of dry steam in the mixture is termed the **quality of the steam.** If we let q represent the quality of the steam, then the latent heat in 1 pound of wet steam equals

$$\frac{qL}{100}$$

and the total heat in 1 pound of wet steam equals

$$h + \frac{qL}{100}$$

37. Steam Tables.—The following table shows the properties of dry saturated steam. More complete tables will be found in Marks and Davis' "Steam Tables" and in the engineering handbooks. Column 1 gives the absolute pressure of the steam in pounds per square inch. Absolute pressure is the pressure shown on the steam gage plus the atmosphere or barometric pressure. For sea-level barometer the atmospheric pressure is 14.7 pounds per square inch. Column 2 gives the corresponding temperature of the steam in degrees Fahrenheit. Column 3 gives the heat of the liquid, and column 4 gives the latent heat. Column 5 gives the total heat of the steam and is the sum of the quantities in columns 3 and 4. Column 6 is the volume of 1 pound of dry saturated steam at the different pressures. Column 7 is the weight of 1 cubic foot of steam at the different pressures.

TABLE X.—PROPERTIES OF SATURATED STEAM[1]

1 Absolute pressure, lb. per sq. in.	2 Temp., deg. F.	3 Heat of the liquid	4 Latent heat of evap.	5 Total heat of the steam	6 Sp. vol., cu. ft. per lb.	7 Density, lb. per cu. ft.
p	t	h	L	H	v	$1/v$
10	193.22	161.1	982.0	1,143.1	38.38	0.02606
11	197.75	165.7	979.2	1,144.9	35.10	0.02849
12	201.96	169.9	976.6	1,146.5	32.36	0.03090
13	205.87	173.8	974.2	1,148.0	30.03	0.03330
14	209.55	177.5	971.9	1,149.4	28.02	0.03569
15	213.00	181.0	969.7	1,150.7	26.27	0.03806
16	216.30	184.4	967.6	1,152.0	24.79	0.04042
17	219.40	187.5	965.6	1,153.1	23.38	0.04279
18	222.40	190.5	963.7	1,154.2	22.16	0.04512
19	225.20	193.4	961.8	1,155.2	21.07	0.04746
20	228.00	196.1	960.0	1,156.2	20.08	0.04980
21	230.60	198.8	958.3	1,157.1	19.18	0.05213
22	233.10	201.3	956.7	1,158.0	18.37	0.05445
23	235.50	203.8	955.1	1,158.8	17.62	0.05676
24	237.80	206.1	953.5	1,159.6	16.93	0.05907
25	240.10	208.4	952.0	1,160.4	16.30	0.0614
30	250.30	218.8	945.1	1,163.9	13.74	0.0728
35	259.30	227.9	938.9	1,166.8	11.89	0.0841
40	267.30	236.1	933.3	1,169.4	10.49	0.0953
45	274.50	243.4	928.2	1,171.6	9.39	0.1065
50	281.00	250.1	923.5	1,173.6	8.51	0.1175
55	287.10	256.3	919.0	1,175.4	7.78	0.1285
60	292.70	262.1	914.9	1,177.0	7.17	0.1394
65	298.00	267.5	911.0	1,178.5	6.65	0.1503
70	302.90	272.6	907.2	1,179.8	6.20	0.1612
75	307.90	277.4	903.7	1,181.1	5.81	0.1721
80	312.00	282.0	900.3	1,182.3	5.47	0.1829
85	316.30	286.3	897.1	1,183.4	5.16	0.1937
90	320.30	290.5	893.9	1,189.4	4.89	0.2044
95	324.10	294.5	890.9	1,185.4	4.65	0.2151
100	327.80	298.3	888.0	1,186.3	4.429	0.2258
105	331.40	302.0	885.2	1,187.2	4.230	0.2365
110	334.80	305.5	882.5	1,188.0	4.047	0.2472
115	338.10	309.0	879.8	1,188.8	3.880	0.2577
120	341.30	312.3	877.2	1,189.6	3.726	0.2683
125	344.40	315.5	874.7	1,190.3	3.583	0.2791
130	347.40	318.6	872.3	1,191.0	3.452	0.2897
135	350.30	321.7	869.9	1,191.6	3.331	0.3002

38. Mechanical Mixtures.—Problems involving the resulting temperature and final condition when various substances at

[1] From MARKS and DAVIS' "Steam Tables and Diagrams."

different temperatures are mixed mechanically are often met with in heating work. They are best treated by first determining the heat in B.t.u. that would be available for use if the temperature of all of the substances were brought to 32°F., and using this heat (positive or negative) to raise (or lower) the total weight of the mixture to its final temperature and condition. Another method of solving such problems is by equating the heat absorbed to the heat rejected and solving for t, the resulting temperature. It is often difficult to decide upon which side of the equation a material should be placed. In such a case a trial calculation should be made, and the temperature determined by the trial will settle this question.

In a mixture of substances which pass through a change of state during the mixture process it is almost necessary to make a trial calculation. Take for example a mixture of steam with other substances. The steam may all be condensed and the resulting water cooled also; the steam may be condensed only; or the steam may be only partially condensed. The equations in each case would be different.

If 1 pound of dry saturated steam at a temperature t_1 is condensed and then the temperature of the condensed steam is lowered to a temperature t_2, the amount of heat H' given off would be

$$H' = L_1 + c(t_1 - t_2)$$

where L_1 is the latent heat corresponding to the temperature t_1 and c is the specific heat of water. If the steam were condensed only, the heat given off would be

$$H' = L_1$$

and the temperature of the mixture is the temperature corresponding to the pressure. If the steam is only partly condensed let q' equal the per cent. of steam condensed. Then

$$H' = \frac{q'L_1}{100}$$

and the temperature of the mixture is the temperature corresponding to the pressure.

The general laws of thermodynamics do not apply in the case of mixtures as the equations become discontinuous.

The general expression for heat absorbed in passing from a solid to a gaseous state may be stated as follows:

Let c_1, c_2, c_3 be the specific heats of the material in the solid, liquid, and gaseous states, respectively. Let w be the weight of the material, t the initial temperature, t_1 the temperature of the melting point, t_2 the temperature of the boiling point, t_3 the final temperature, H_f the heat of liquefaction, and L the heat of vaporization. Then

$$H' = w[c_1(t_1 - t) + H_f + c_2(t_2 - t_1) + L + c_3(t_3 - t_2)]$$

Example.—Find the final temperature and condition of the mixture after mixing 10 pounds of ice at 20°, 20 pounds of water at 50° and 2 pounds of steam at atmospheric pressure. Mixture takes place at the pressure of the steam. The specific heat of ice may be taken as 0.5 and the heat of liquefaction as 144 B.t.u.

First Method

Solution.—

Heat to raise ice to 32° $= 10 \times 0.5(32 - 20)$	$= 60.0$
Heat to melt ice $= 10 \times 144$	$= 1440$
Total heat necessary to change the ice to water at 32°	$= 1500$ B.t.u.
Heat given up by water when temperature is lowered to 32° $= 20 \times (50 - 32)$	$= 360.0$
Heat in steam above 32° (from tables) $= 2 \times 1150.3$	$= 2300.6$
Total heat given up in lowering water and steam to 32°	$= 2660.6$ B.t.u.
Heat available for use $= 2660.8 - 1500$	$= 1160.6$ B.t.u.
Degrees this heat will raise the mixture $1160.6 \div 32$	$= 36.3$

$\therefore$ Final temperature of mixture $= 36.3 + 32 = 68.3$°F.

Ans. 32 pounds water at 68.3°F.

Second Method

Assume that the steam is all condensed and that the final temperature of the mixture is t. Then the heat necessary to raise the ice to the melting point equals

$$10 \times 0.5(32 - 20)$$

The heat necessary to melt the ice equals 10×144; the heat necessary to raise the melted ice to the temperature of the mixture equals $10(t - 32)$; the heat necessary to raise the water to the temperature of the mixture equals $20(t - 50)$; the heat given up by the steam in changing to water at the temperature of the boiling point equals 2×970.4, and the heat given up by the condensed steam when its temperature is lowered to the temperature of the mixture equals $2(212 - t)$.

Combining the preceding parts into one equation, we have

$$10 \times 0.5(32-20) + 10 \times 144 + 10(t-32) + 20(t-50) = 2 \times 970.4 + 2(212-t)$$

$$60 + 1440 + 10t - 320 + 20t - 1000 = 1940.8 + 424 - 2t$$

$$32t = 2184.8$$
$$t = 68.3^\circ$$

Since t is less than the temperature of the boiling point corresponding to the pressure at which the mixture takes place, all the steam is condensed. *Ans.* 32 pounds water at 68.3°F.

Example.—Find the resulting temperature and condition after mixing 10 pounds of ice at 20°, 20 pounds of water at 50°, 40 pounds of air at 82°, and 20 pounds of steam at 100 pounds gage pressure and containing 2 per cent. moisture. Mixture takes place at the pressure of the steam.

First Method

Solution.—

$10 \times 0.5(32 - 20)$	=	60	
10×144	=	1440	
		1500 B.t.u.	= heat to raise ice to water at 32°.
$20 \times (50 - 32)$	=	360	
$40 \times 0.2415(82 - 32)$	=	483	
$20(308.8 + 0.98 \times 880.0)$	=	23,424	
		24,267 B.t.u.	= heat given up by air, water, and steam.
		1,500	
		22,767 B.t.u.	= heat available.
$40 \times 0.2415(337.9 - 32)$	=	2,955 B.t.u.	= heat to raise air to 337.9°.
		19,812 B.t.u.	= heat available to raise the water.
50×308.8	=	15,440 B.t.u.	= heat to raise water to 337.9°.
		4,372 B.t.u.	= heat available to evaporate water.

$$\frac{4372}{880.0} = 4.97 \text{ pounds steam.}$$

Ans. 40.00 pounds air, 45.03 pounds water, 4.97 pounds dry saturated steam, at 337.9°.

Second Method

Assume the steam to be all condensed and let the temperature of the mixture be t°. Equating the heat gained by the ice, water, and air, and the heat lost by the steam, we have

$$10 \times 0.5(32 - 20) + 10 \times 144 + 10(t - 32) + 20(t - 50) + 40 \times 0.2415(t - 82) = 20 \times 0.98 \times 880.0 + 20(337.9 - t)$$

$$60 + 1440 + 10t - 320 + 20t - 1000 + 9.7t - 792 = 17{,}248 + 6758 - 20t$$

$$59.5t = 24,618$$
$$t = 413.7°\text{F}.$$

This result is of course absurd, as the temperature of the mixture cannot be higher than the temperature of the boiling point corresponding to the pressure at which the mixture takes place. Therefore, our assumption that all the steam is condensed must be wrong, and we know that part of it remains in the form of steam, and hence the temperature of the mixture is equal to the temperature of the boiling point corresponding to the pressure at which the substances are mixed.

Then, substituting for t its value, and letting x represent the number of pounds of steam condensed, we have

$$10 \times 0.5(32 - 20) + 10 \times 144 + 10(337.9 - 32) + 20(337.9 - 50) + 40 \times 0.2415(337.9 - 82) = 880.0x$$

$$60 + 1440 + 3059 + 5758 + 2472 = 880.0x$$
$$880.0x = 12,789$$
$$x = 14.53 \text{ pounds condensed.}$$

$$20 \times 0.98 = 19.6 \text{ pounds} = \text{original weight of dry steam.}$$

Ans. 40 pounds air
$10 + 20 + (20 - 19.6) + 14.53 = 44.93$ pounds water
$19.6 - 14.53 = 5.07$ pounds dry saturated steam
} at 337.9°.

The difference between the results obtained in these two methods of working this problem is due to the fact that in the first method we took account of the variation in the specific heat of water by using the heat of the liquid, h, from the tables, in place of $(t - 32)$ wherever possible, while in the second method we assumed this specific heat to be constant and equal to 1.

Example.—Find the resulting temperature and condition after mixing 10 pounds of ice at 20°, 20 pounds of water at 50°, and 30 pounds of steam at 100 pounds pressure and 400° temperature. Mixture takes place at 25 pounds pressure.

FIRST METHOD

Solution.—

$10 \times 0.5(32 - 20)$	= 60	
10×144	= 1,440	
	1,500 B.t.u.	= heat to raise ice to water at 32°.
$20 \times (50 - 32)$	= 360	
$30 \times 0.53(400 - 337.9)$	= 987	
30×1188.8	= 35,664	
	37,013 B.t.u.	= heat given up by water and steam.
	1,500	
	35,513 B.t.u.	= heat available.
60×235.6	= 14,136 B.t.u.	= heat to raise water to 266.8°.
	21,377 B.t.u.	= heat available to evaporate water.

$$\frac{21,377}{933.6} = 22.89 \text{ pounds steam.}$$

Ans. $\left.\begin{array}{l}\text{37.11 pounds water}\\ \text{22.89 pounds dry saturated steam}\end{array}\right\}$ at 266.8°F.

Second Method

Assume the steam to be all condensed and let the temperature of the mixture be $t°$. Then

$$10 \times 0.5(32 - 20) + 10 \times 144 + 10(t - 32) + 20(t - 50) = 30 \times 0.53 (400 - 337.9) + 30 \times 880.0 + 30(337.9 - t)$$

$$60 + 1440 + 10t - 320 + 20t - 1000 = 987 + 26,400 + 10,137 - 30t$$

$$60t = 37,344$$
$$t = 622.4°$$

This result is, of course, impossible and we see at once that only part of the steam is condensed, and that the temperature of the mixture must be that of the boiling point corresponding to the pressure at which the mixture takes place.

This problem differs from the previous ones in that the pressure of the mixture is different from the original steam pressure, and we must proceed in a slightly different manner.

Assume for the moment that the steam has all been condensed and that we have 60 pounds of water at 622.4°F. Then assume that the temperature of the water is dropped to the temperature of the boiling point (266.8°) corresponding to the pressure (25 pounds) at which the mixture is made. Each pound will give up, approximately (622.4 − 266.8) B.t.u. This heat can then be used to re-evaporate part of the water. Therefore, since the latent heat corresponding to 25 pounds is 933.6, we have

$$\frac{60(622.4 - 266.8)}{933.6} = \frac{60 \times 355.6}{933.6} = \frac{21,330}{933.6} = 22.85 \text{ pounds re-evaporated.}$$

Ans. $\left.\begin{array}{l}\text{37.15 pounds water}\\ \text{22.85 pounds dry saturated steam}\end{array}\right\}$ at 266.8°F.

Problems

1. Required the temperature after mixing 3 pounds of water at 100°F., 10 pounds of alcohol at 40°F., and 20 pounds of mercury at 60°F.

2. Required the temperature and condition after mixing 5 pounds of ice at 10°F. with 12 pounds of water at 60°F.[1]

3. Required the temperature and condition after mixing 10 pounds of ice at 15°F. with 1 pound of water at 212°F.

4. Required the temperature and condition of the mixture after mixing 5 pounds of steam at 212°F. with 20 pounds of water at 60°F.

5. One pound of ice[2] at 32° is mixed with 10 pounds of water at 50° and

[1] Specific heat of ice equals 0.5.

[2] Latent heat of fusion of ice = 144 B.t.u.'s.

20 pounds of steam at 212°. What is the temperature and condition of the resulting mixture?

6. Ten pounds of steam at 212° are mixed with 50 pounds of water at 60° and 2 pounds of ice at 32°. What will be the resulting temperature and condition of the mixture?

7. Ten pounds of steam at atmospheric pressure, 5 pounds of water at 50° and 10 pounds of ice at 32° are mixed together. (*a*) What will be the resulting temperature of the mixture? (*b*) What will the condition of the mixture be? (*c*) If the steam is not all condensed, determine what per cent. of the steam will be condensed.

8. Five pounds of steam at atmospheric pressure, 10 pounds of water at 60°, and 2 pounds of ice at 20° are mixed at atmospheric pressure. What will be the resulting temperature?

9. Ten pounds of ice at 10°, 20 pounds of water at 60° and 5 pounds of steam at atmospheric pressure are mixed at atmospheric pressure. Find the resulting temperature and condition of the mixture.

10. Twenty pounds of steam at atmospheric pressure, 10 pounds of water at 60° and 50 pounds of air at 100° are mixed together at the pressure of the steam. (*a*) What will be the resulting temperature? (*b*) If the steam is not all condensed, determine what per cent. of the steam will be condensed.

11. A mixture is made of 10 pounds of steam at atmospheric pressure, 5 pounds of ice at 20°, 10 pounds of water at 50°, 30 pounds of air at 60°. (*a*) What will be the temperature of the resulting mixture? (*b*) What will be the percentages by weight of air, steam, and water in the mixture?

12. What would be the resulting temperature and condition of a mixture of 10 pounds of water at 40°, 20 pounds of water at 60°, and 8 pounds of steam at 5 pounds pressure? Mixture takes place at 5 pounds pressure.

13. Ten pounds of steam at 5 pounds pressure, 1 pound of ice at 32°, and 20 pounds of water at 60° are mixed at 5 pounds pressure. What will be the temperature and condition of the resulting mixture?

14. Five pounds of ice at 5°, 10 pounds of water at 50°, 20 pounds of air at 80°, and 5 pounds of steam at 20 pounds pressure are mixed at the pressure of the steam. Find the resulting temperature and condition of the mixture.

15. Required the temperature and condition of the mixture after mixing 10 pounds of steam at a pressure of 30 pounds absolute and a temperature of 250.3°F., 2 pounds of ice at 10°F., and 20 pounds of water at 40°F. Mixture takes place at the pressure of the steam.

16. Fifty pounds of air at 100°, 10 pounds of steam at atmospheric pressure, and 10 pounds of water at 60° are mixed at atmospheric pressure. What is the temperature of the mixture and how much steam is condensed?

17. Required the final temperature and condition after mixing at the pressure of the air 100 pounds of air at a temperature of 500° and a pressure of 100 pounds absolute, and 2 pounds of steam at 100 pounds absolute having a quality of 98 per cent.

18. Five pounds of steam at 5 pounds gage pressure are mixed at atmospheric pressure with 10 pounds of water at 60°. What is the temperature and condition of the resulting mixture?

19. Thirty pounds of water at 60°, 10 pounds of steam at 115 pounds

absolute and a temperature of 400°F., and 10 pounds of ice at 20° are mixed at atmospheric pressure. What will the resulting temperature be? What is the condition of the mixture?

20. Ten pounds of ice at 20°F., 18 pounds of water at 80°, and 10 pounds steam at 75 pounds pressure and 90 per cent. quality, are mixed at atmospheric pressure. What is the resulting temperature and condition of the mixture?

21. Two pounds of steam at 150 pounds absolute and a temperature of 400°, 5 pounds of ice at 22°, and 10 pounds of water at 60° are mixed at atmospheric pressure. Find the final temperature and condition of mixture.

22. Required the final temperature and condition after mixing at atmospheric pressure 3 pounds of ice at 22° and 3 pounds of steam at 100 pounds pressure and containing 2 per cent. moisture.

23. Find the resulting temperature and condition of a mixture of 10 pounds of steam at 150 pounds absolute and a temperature of 400°F., 10 pounds of water at 60°F., and 50 pounds of air at 112°F. Mixture takes place at atmospheric pressure.

24. Five pounds of ice at 0°, 20 pounds of water at 75°, and 15 pounds of steam at 50 pounds absolute and 95 per cent. quality are mixed at 20 pounds absolute. What is the resulting temperature and condition of the mixture?

25. How many pounds of water will 10 pounds of dry steam heat from 50° to 150° if the steam pressure is 100 pounds gage?

26. If 10 pounds of steam at 100 pounds gage raised 93 pounds of water from 50° to 140°, what per cent. of moisture is in the steam, radiation being zero?

27. A pound of steam and water occupies 3 cubic feet at 110 pounds absolute pressure. What is the quality of the steam?

CHAPTER V

RADIATORS

39. Classification.—In a steam or hot-water heating system the conveying medium absorbs heat at the boiler and then flows to the radiators whose function is to transmit the heat to the air, walls, etc. of the room. There are several forms of radiation, the proper one to be used in any particular case depending upon the nature and use of the building.

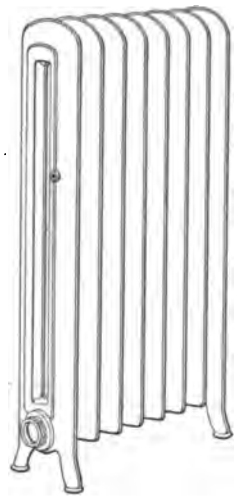
The selection of radiators of the proper size for each room in the building is very important. If the radiators are too small it will be impossible in the coldest weather to warm the building to the required temperature within a reasonable time, if at all. On the other hand, the installation of radiators of too large a size adds unnecessarily to the cost of the heating system, and tends to cause the rooms to be overheated during a large part of the time. In order to compute intelligently the amount of radiating surface required, it is necessary to study the various forms of radiation and the factors affecting the rate of heat transmission from each.

Radiators may be divided into three classes: (*a*) **direct** radiators, (*b*) **indirect** radiators, and (*c*) **semi-indirect** radiators. Direct radiators, as explained in Chapter III, are located in the rooms to be heated, while indirect radiators are located elsewhere and a current of air conveys the heat from them to the rooms. Semi-indirect radiators are a combination of the other two forms, the radiators being installed in the rooms but delivering a large proportion of their heat output by means of a current of air which passes through them.

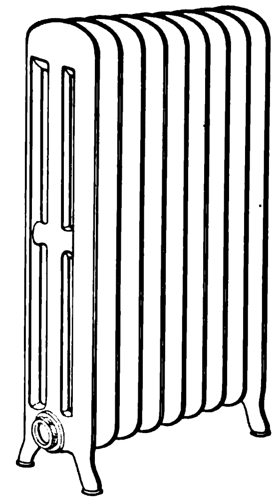
40. Direct Cast-iron Radiators.—Direct radiators are made of cast iron, pressed iron, and wrought iron or steel pipe, the cast-iron radiator being by far the most widely used. It is composed of several sections cast separately and assembled, the number of sections being fixed by the amount of surface required. The sections are made in several different widths and heights so that for a radiator of a given surface, a wide range of shapes and

sizes is available. The wider sections are divided through most of their length by vertical slots into from two to six segments or

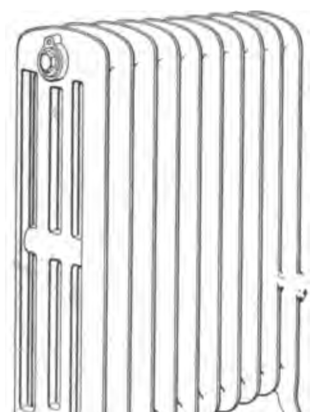
Single Column Radiator



Two Column Radiator



Three Column Radiator



Four Column Radiator

Window Type

Fig. 8

"columns." The standard heights vary from 15 to 45 inches but the 38-inch height is the one most often used. In Fig. 8 are

shown several forms of cast-iron radiators. Radiators are finished in several designs to harmonize with room decorations.

In general appearance the form of radiator used for steam is quite similar to that used for water. The two designs are fundamentally different, however, in that the sections of the steam radiator are joined together at the bottom only, while those in a hot-water radiator are connected at both top and bottom. Hot-water radiation may be used for steam but steam radiation could not be satisfactorily used in a hot-water system because air would become trapped in the top of each of the sections, preventing the water from filling them.

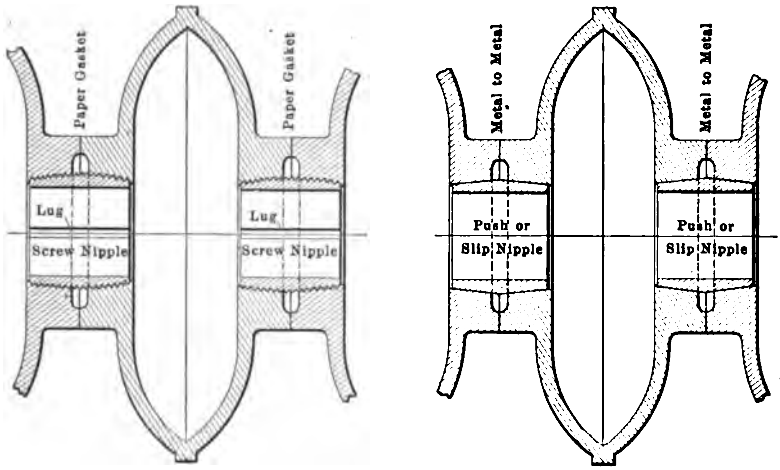


FIG. 9.—Methods of assembling cast-iron radiators.

The sections are joined by means of nipples. One method is to use a smooth tapered "push nipple," fitting into tapered holes in the adjacent sections. Draw-bolts extending the full length of the radiator are used to force the joints to a tight fit. Another method is to use nipples threaded with "right and left" threads. These nipples are cast with internal lugs and are turned up by means of a special wrench. The two methods of assembling are shown in Fig. 9.

Cast-iron radiators are usually given a hydraulic pressure test at the factory of about 120 pounds per square inch. They are therefore suitable for working pressures approaching this figure but are seldom subjected to any such pressure except in the case of hot-water systems in tall buildings where the hydrostatic

head is high. The weight of cast-iron radiators averages about 7 pounds per square foot of surface and the internal volume is about 30 cubic inches per square foot of surface. This internal volume is largely fixed by the requirements of manufacture, the only stipulation from an engineering standpoint being that the passages must not be so small as to restrict the flow of the water or steam.

Cast-iron radiation is also furnished in the form of "wall radiators" as illustrated in Fig. 10. This type of radiation is so

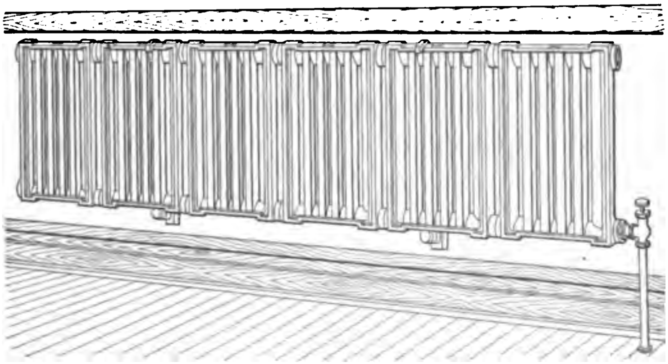


FIG. 10.—Wall radiator.

proportioned that it takes up very little lateral space and is intended to be hung from brackets. It is well adapted for use in factory buildings.

The rated external surface of radiators of various widths and heights is given in Table XI in square feet of surface per section.

TABLE XI.—HEATING SURFACE PER SECTION—CAST-IRON RADIATION

Height, inches	One-column	Two-column	Three-column	Four-column	Six-column or "window" pattern
45	...	5	6	10	
38	3	4	5	8	
32	2½	3⅓	4½	6½	
26	2	2⅔	3¾	5	
23	1⅔	2⅓	...	...	
22	...	2¼	3	4	
20	1½	2	...	...	5
18	...	...	2¼	3	...
16	...	...	...	...	3¾
15	...	1½	...	...	...
14	...	...	...	...	...
13	...	...	...	...	3

WALL RADIATORS

Size of section, inches (approx.)	Heating surface, square feet
14 by 16	5
14 by 22	7
14 by 29	9

It should be noted that the height of a radiator is taken as the **total height above the floor** for radiators having legs of standard height. The rated surface given in the table does not correspond exactly with the actual surface, but the difference may be neglected as the heat transmission from radiators is usually given in terms of rated surface.

41. Radiator Tappings.—The end sections of cast-iron radiators are usually tapped for a 2-inch pipe thread and furnished with bushings having openings whose size depends on the size of the radiator. The sizes of the reduced openings for radiators intended for use with different systems of piping are as follows:

TABLE XII.—RADIATOR TAPPINGS

Size of radiator, square feet	Inches
Single-pipe Work	
Up to 24	1
24 to 60	1¼
60 to 100	1½
Above 100	2
Two-pipe Work, supply and return	
Up to 48	1 by ¾
48 to 96	1¼ by 1
Above 96	1½ by 1¼
Water radiators, supply and return	
Up to 40	1 by 1
40 to 72	1¼ by 1¼
Above 72	1½ by 1½

For vapor systems supply, ¾ inch, return, ½ inch. Air valve tapping, ⅛ inch on all radiators.

42. Pressed-metal Radiators.—In recent years radiators made of pressed metal have been introduced and are now sometimes used. Fig. 11 illustrates the appearance of one design of this form of radiator, and Fig. 12 is a cross-section. The sections are made of two sheets of metal pressed to shape and welded at the edges. In other designs the joint is a lapped seam. A special alloy- or soft steel selected for its non-corroding qualities is used. The radiator is assembled by welding the sectors

together or by joining them with lapped seams. Pressed-metal radiators are made in a variety of sizes corresponding to those of cast-iron radiation. The sections are very narrow and occupy much less space than do cast-iron radiators of equal surface. The weight per square foot of surface is also much less than that of cast-iron radiation, averaging about 2 pounds. The cost is about the same as that of ordinary cast-iron radiation. The radiating surface of pressed-metal sections of various heights and widths is given in Table XIII. Because of its light weight this

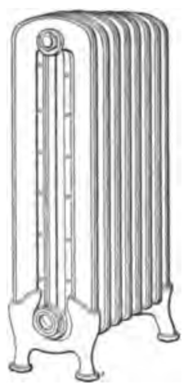


FIG. 11.—Pressed metal radiator.

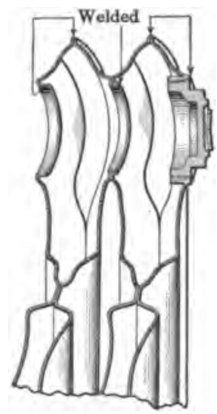


FIG. 12.—Section of pressed metal radiator.

form of radiation is especially suitable for hanging on wall brackets.

TABLE XIII.—PRESSED-METAL RADIATION, SQUARE FEET OF SURFACE PER SECTION

Height of radiator, inches	Width of section, inches	
	4½	8¼
45	...	6
38	3	5
32	2½	4½
26	2	3¾
22	1⅔	3
18	1⅓	2¼
14	1	...

43. Pipe Radiation.—In factories and other industrial buildings radiators built of pipe are often used and are a very satisfactory

form of radiation. These pipe coils usually consist of a pair of cast-iron headers connected by four or more pipes of either 1 inch or $1\frac{1}{4}$ inches diameter. Pipe coils are usually made in the mitre form as shown in Fig. 13. The vertical lengths of pipe provide sufficient flexibility to allow the longer horizontal members to expand freely. Some such provision is essential. The openings in one of the headers or the elbows are tapped

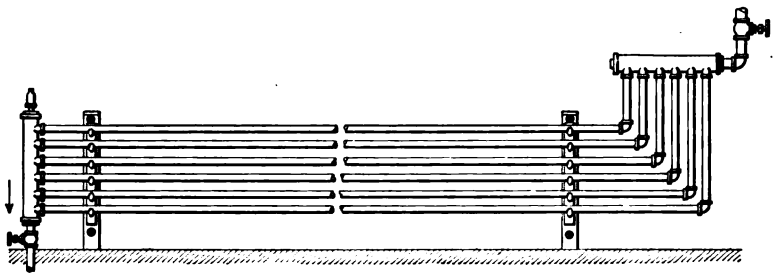


Fig. 13.—Mitre pipe coil.

with a left-hand thread so that the coil can be readily assembled. Pipe coils of the form shown in Fig. 14 are also sometimes used, especially in hot-water work.

Radiators were formerly made of vertical pipes screwed into a cast-iron base. This form of radiation is little used at present.

44. Heat Transmission from Radiators.—Heat flows from the water or steam in a radiator into and through the metal wall

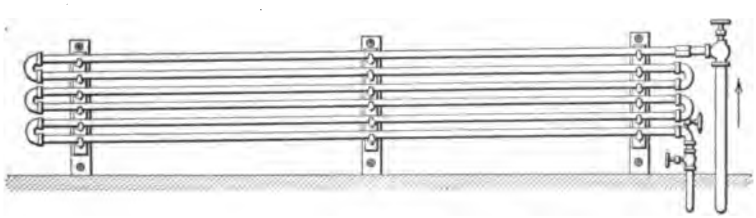


Fig. 14.—Continuous pipe coil.

and is transmitted from the outer surface partly by radiation and partly by convection. The resistance to heat flow offered by the walls of the radiator is so slight that the temperature of the outer surface is practically the same as that of the water or steam. It is very difficult to measure accurately the portions of the total amount of heat which are transmitted by radiation and by convection. Rough tests, however, indicate that about one-half of the total amount is given off in each manner. The total

amount of heat transmitted per square foot of radiating surface is affected by several factors, such as the temperature difference between the radiating surface and the surrounding air, the nature of the surface, the height and shape of the radiator, and the location of the radiator in the room.

45. Effect of Shape of Surface.—The **form or shape of the radiator** has a marked effect on the heat transmission, affecting both the amount radiated and that given off by convection. A greater amount of heat per square foot of surface is given off by radiation from a pipe coil or a single-column radiator than from a radiator of a wider pattern. This can be clearly understood from a study of Fig. 15 which represents horizontal cross-sections of a single-column and a three-column radiator.

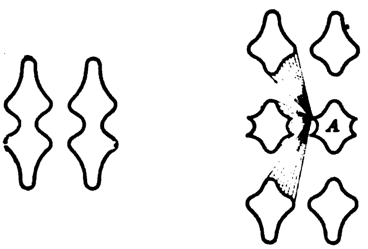


FIG. 15.

The rays of heat from points on the single-column radiator can travel in nearly any direction without interruption, while the rays emanating from many points such as A, on the surface of the inner columns of the three-column radiator, are largely intercepted by the other portions of the radiator.

The transmission of heat by convection is dependent upon the difference in temperature between the surface of the radiator and the air. The upper part of a radiator will transmit less heat per square foot by convection than will the lower part because of the increase in the temperature of the air as it ascends along the surface. Hence **the average heat transmission per square foot is greater for short than for tall radiators,** and for the same reason a radiator or pipe coil laid on its side will give off more heat than when in a vertical position.

46. Effect of Painting.—The effect of the decorative painting on the heat transmission is sometimes considerable. Experiments made at the University of Michigan indicate that (*a*) if several paints of different kinds are applied successively the effect on the heat transmission **is due entirely to the final coat,** and (*b*) the aluminum or bronze paints have the greatest effect, reducing the heat transmission almost 25 per cent. in some cases. The relative effect of different kinds of paints is given in Table XIV.

TABLE XIV.—RELATIVE EFFECT OF RADIATOR PAINTS

Kind of paint	Relative transmission
Bare iron surface	1.000
Copper bronze	0.760
Aluminum bronze	0.752
Snow-white enamel	1.010
No-luster green enamel	0.956
Terra-cotta enamel	1.038
Maroon glass Japan	0.997
White lead paint	0.987
White zinc paint	1.010

47. Coefficients of Heat Transmission.—The amount of heat transmitted from a radiator may be represented by the expression,

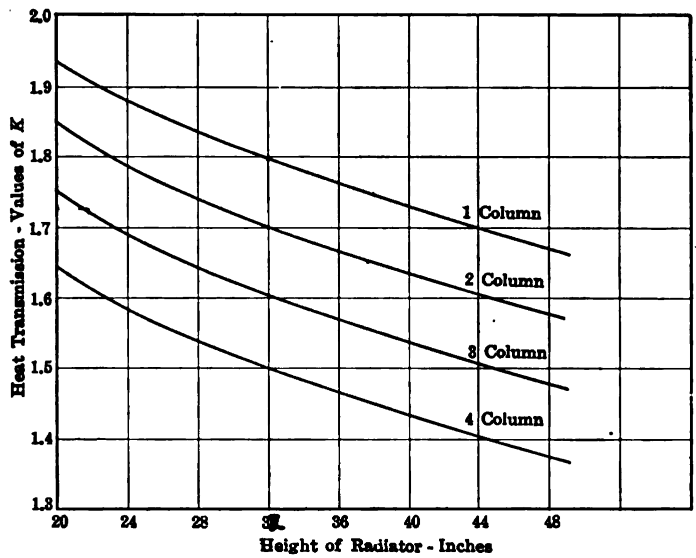


FIG. 16.—Coefficient of heat transmission from radiators.

$H = SK(t_s - t_r)$ in which

S = the area of the radiating surface in square feet.

K = the coefficient of heat transmission in B.t.u. per square foot per hour per degree difference between radiator and room temperature.

t_s = temperature of the steam or water in the radiator.

t_r = room temperature.

The values of K, the coefficient of heat transmission for ordinary cast-iron radiation of various heights and widths, is given by the curves in Fig. 16 which are based on the results of

recent experiments. For other forms of radiation the values of K given in Table XV may be taken as average figures.

TABLE XV.—COEFFICIENT OF HEAT TRANSMISSION FROM RADIATORS

	K B.t.u. per square foot per hour per degree difference in temperature
Cast iron, height 38 inches:	
One-column	1.75
Two-column	1.65
Three-column	1.55
Four-column	1.45
Wall Coil:	
Heating surface 5 square feet, long side vertical	1.92
Heating surface 5 square feet, long side horizontal	2.11
Heating surface 7 square feet, long side vertical	1.70
Heating surface 7 square feet, long side horizontal	1.92
Heating surface 9 square feet, long side vertical	1.77
Heating surface 9 square feet, long side horizontal	1.98
Pipe Coil:	
Single horizontal pipe	2.65
Single vertical pipe	2.55
Pipe coil 4 pipes high	2.48
Pipe coil 6 pipes high	2.30
Pipe coil 9 pipes high	2.12

This data is based on a temperature difference between the radiator and the air of about 150° which represents ordinary conditions. The rate of heat transmission increases slightly with an increase in the temperature difference. In Table XVI are given the results of a test on a 38-inch two-column radiator

TABLE XVI.—COEFFICIENT OF HEAT TRANSMISSION FOR VARYING TEMPERATURE DIFFERENCE BETWEEN THE RADIATOR AND ROOM

Difference in temperature, degrees	Coefficient of heat transmission, K
80	1.560
100	1.580
120	1.615
140	1.645
150	1.650
160	1.675
170	1.690
180	1.705
190	1.720

showing this change in the value of K with an increasing temperature difference.

For ordinary conditions, that is, when the system is to be designed for a steam pressure of from 1 to 5 pounds and the room temperature is 70° or thereabouts, there will be no necessity for considering the change in the heat transmission with varying temperature differences. Occasionally, however, there are

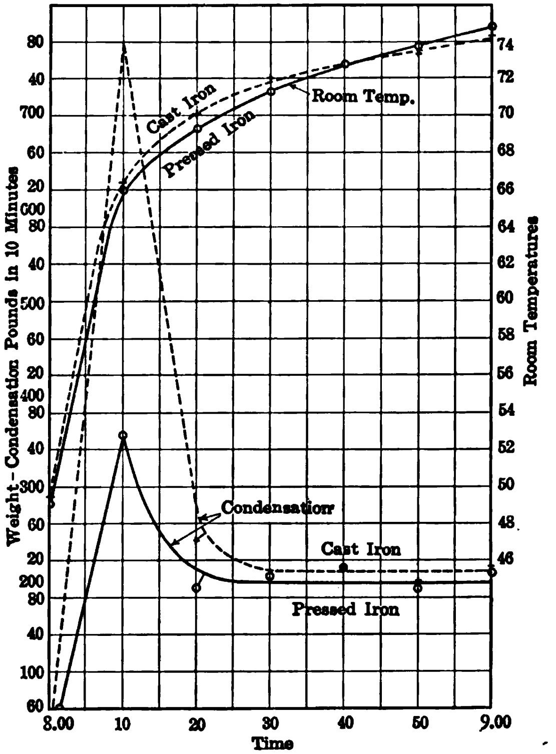


FIG. 17.—Result of a comparative test of a cast iron and a pressed iron radiato.

conditions such as in drying rooms and similar places that are to be kept at a very high temperature where it will make an appreciable difference in the amount of radiation required. In some vacuum systems, also, where a very high vacuum is to be carried even in the coldest weather, it is desirable to take this factor into consideration.

The heat transmission from pressed-metal radiation is practically the same as that from cast iron. This is illustrated in Fig.

17 which shows the results of a test[1] to determine the relative performance of the two forms of radiation under the same conditions. A radiator of each kind was placed in either of two similar rooms and the condension formed in each radiator was weighed at 10-minute intervals and the room temperatures were measured. While the rate at which the room was warmed was nearly the same in both cases it will be noted that in the case of the cast-iron radiator the initial condensation of steam is considerably greater.

48. The Location of Radiators.—The location of the radiators is of considerable importance from several standpoints. Unless

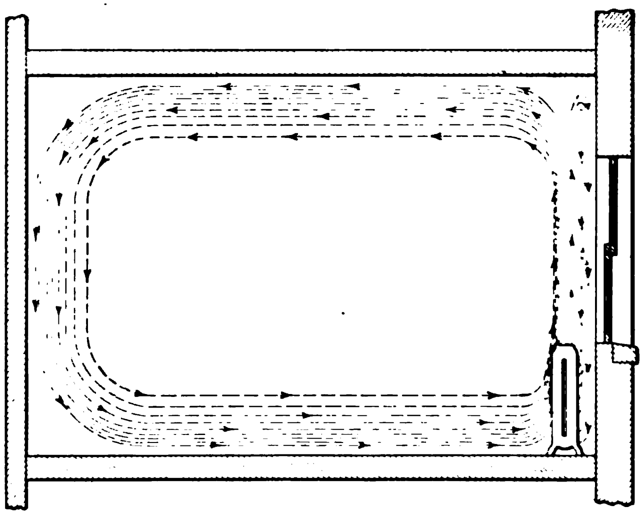


FIG. 18.—Effect of locating radiator beneath window.

there are columns or other permanent structures in the interior of the room, it is necessary, at the outset, to place the radiators around the walls. The piping is also simplified by placing the radiators near the walls. If the radiators are placed against an interior wall there is a tendency for uncomfortable draughts to be formed by the cooling effect of the windows and outer wall tending to form a downdraught on one side of the room, together with the effect of the upward movement created by the radiator on the other side. If the radiator is placed under the window,

[1] See "Coefficient of Heat Transmission in a Pressed-Metal Radiator" by JOHN R. ALLEN, *Trans.* A. S. H. & V. E., 1914.

the current of air rising from the radiator will counteract this tendency and will produce an air movement as illustrated in Fig. 18. The downward current caused by the cooling effect of the window causes a secondary circulation of the air between the radiator and the window. **The location of the radiators beneath the windows if possible is, on the whole, the most desirable.** Recent tests[1] have indicated that the transmission of heat is slightly greater when the radiators are located in other positions, but the slight gain in effectiveness is greatly overbalanced by the other considerations noted above.

Radiators are often located under seats and shelves or behind grilles of various designs, the object being either to conceal the radiator or to conserve space. The heat transmission from the radiator is usually decreased by such enclosures, because of the restriction imposed on the circulation of the air through the radiator. Where it is necessary to place a radiator in such a location, an addition of from 10 to 30 per cent. should be made to its heating surface according to the degree to which the circulation is retarded by the enclosure.

49. Proportioning Radiation.—The heat loss from the various rooms of a building having been calculated by the methods given in Chapter II, it is then necessary to determine the amount of radiating surface which will be required to supply the heat losses. It is necessary first to know the temperature of the steam or water in the radiator. If steam is the heat carrying medium the temperature will be that corresponding to the pressure to be carried. In many heating systems it is possible to carry a pressure of at least 5 pounds when necessary and for such systems the radiation is commonly figured on the basis of this pressure. If, however, special conditions require that a lower pressure be used the temperature of the steam which is assumed should be that corresponding to the pressure. Some types of vapor heating systems are designed to operate at nearly atmospheric pressure, and the radiation is consequently figured for 212°. If hot water is used the temperature will range between 160° and 200°. The factors affecting the temperatures carried in hot-water systems will be discussed later.

The type of radiation and the height must next be selected from

[1] See report of Committee on Best Position of a Radiator, *Trans.* A. S. H. & V. E., 1916.

a consideration of the nature of the building and of the space available. From the chart in Fig. 16 or from Table XV the heat transmission per square foot of surface for the type of radiation selected can be found and the total surface necessary to transmit the heat required can then be computed. For example, consider that the room shown in Fig. 7, page 23, is to be heated by a heating system which is to operate at a pressure of 2 pounds. The heat loss from the room was found by the B.t.u. method to be 8696 B.t.u. per hour with room temperature 70°. Assume that 38-inch, two-column radiation is to be used. The temperature of steam at 2 pounds pressure is 218.2 and the difference in temperature between the steam and the air is 218.2° − 70° or 148.2°. From the chart in Fig. 16 we see that the value of K for 38-inch, two-column radiation is 1.65. For a temperature difference of 148.2° the heat transmission would be 244 B.t.u. per square foot per hour. Dividing 8696 by this figure we find that 35.6 square feet of radiation would be required. Since 38-inch, two-column radiation contains 4 square feet of surface per section, a radiator of nine sections would be used.

50. Approximate Rules for Calculating Radiation.—The method outlined above should be followed when accurate results are necessary or when the conditions are exceptional. For rough calculations the average rate of heat transmission per degree difference in temperature per hour may be assumed to be 1.65 B.t.u. If the steam pressure is assumed to be 5 pounds the temperature will be 227° and the temperature difference between the radiator and the room, assuming the room temperature at 70°, would be 157°. The heat transmission per square foot of radiation per hour would then be $1.65 \times 157 = 259$ B.t.u.

Having computed the heat loss by either of the methods given in Chapter II the radiation required can be approximately determined by dividing the computed heat loss by 259.

51. Checking a Contractor's Guarantee.—The case often arises in which a contractor has guaranteed that the heating system as installed is capable of heating the building to 70° in zero weather and it is desired to prove that this is true without waiting for extremely cold weather. By means of the following method it is possible to determine the temperature to which the building must be heated in the warmer weather if the heating system is capable of heating it to the guaranteed temperature in the coldest weather.

Let t_1 = temperature of outside air under contract conditions, usually 0°.

t_2 = temperature of air in building under contract conditions.

t_3 = temperature of steam in radiator at pressure specified. Test made with steam at same pressure.

t_4 = temperature of outside air during test.

t_5 = inside temperature to be maintained during test if system fulfills guarantee.

h = computed heat loss from building per degree difference in temperature.

The heat loss from the building under conditions specified in guarantee would be

$$h(t_2 - t_1) \tag{1}$$

The heat loss from the building under test conditions is

$$h(t_5 - t_4) \tag{2}$$

The heat loss from the radiators under contract conditions would be

$$K(t_3 - t_2) \tag{3}$$

in which K is the coefficient of heat transmission from the radiator. The heat transmission from the radiator under test conditions is

$$K(t_3 - t_5) \tag{4}$$

Then the quantity (1) must be equal to the quantity (3) and the quantity (2) must be equal to (4), hence

$$h = \frac{K(t_3 - t_2)}{(t_2 - t_1)} \tag{5}$$

and

$$h = \frac{K(t_3 - t_5)}{(t_5 - t_4)} \tag{6}$$

Equating the right-hand members of equations (5) and (6), we have

$$\frac{t_3 - t_2}{t_2 - t_1} = \frac{t_3 - t_5}{t_5 - t_4} \tag{7}$$

Assuming $t_1 = 0°$, $t_2 = 70°$, and $t_3 = 228°$, the temperature corresponding to 5 pounds steam pressure, and solving for t_5 we have

$$t_5 = 0.695t_4 + 70 \tag{8}$$

The following table has been computed from equation (8) and shows the room temperature, for different outside temperatures existing during the test, which must be maintained to fulfill a guarantee which specifies the temperatures and steam pressure given above. For other conditions equation (7) must be solved for t_5.

TABLE XVII.—ROOM TEMPERATURE FOR VARIOUS OUTSIDE TEMPERATURES

Outside temperature during test	Room temperature, two-column radiation	Room temperature, three-column radiation
−30	52.0	53.0
−20	58.0	59.0
−10	64.0	64.0
0	70.0	70.0
10	77.5	75.0
20	83.0	83.0
30	90.0	89.0
40	97.0	95.0
50	103.5	105.5
60	110.0	108.0
70	117.0	115.0
80	123.5	121.5
90	130.0	128.0
100	137.0	134.5

52. Indirect Radiators.—Indirect radiators are so named because they are located outside of the room to be heated and the heat is conveyed from the radiator to the room by a current of air. Indirect radiators are of two classes: **gravity indirects,** in which the circulation of the air over the radiating surface is produced by the difference in weight of the heated and unheated columns of air, and **fan coils,** over which the air is forced by a fan. Only the former will be considered here, the various types of fan systems being discussed in Chapter XV.

There are two reasons for the use of gravity indirect radiators. Their chief advantage is that they can be arranged to introduce fresh air from outside and they are therefore desirable from a standpoint of ventilation. Another advantage is that the radiators are out of sight, which is desirable in any room or apartment where appearance is an important factor. It is seldom that indirect radiators are installed throughout an entire building because of the increased cost both of installation and operation as compared with direct radiation. In a residence, indirect

radiation is often installed in the living rooms where ventilation is most desired and where the appearance of the radiators would be objectionable, and direct radiation is used in the bedrooms, halls, etc. The increased operating cost where indirect radiation is used is due to the fact that the large quantities of air which are brought in from outside must be heated up to room temperature or above.

53. Forms of Indirect Radiation.—As indirect radiators are concealed, their appearance is not an important factor and they are therefore designed and installed from a standpoint of effectiveness rather than appearance. Since the resistance to heat transmission between the outer surface of the radiator and the air is greater than that from the steam or water to the inside surface of the radiator wall, it is desirable to make the external

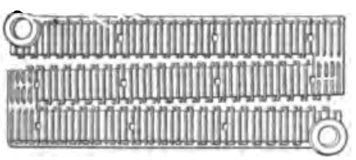


Fig. 19*a*.

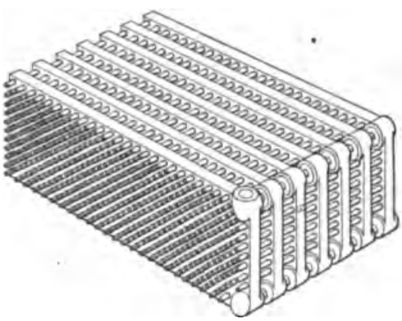


Fig. 19*b*.

Forms of indirect radiators.

surface of greater area than the internal. This is accomplished by adding projections in the form of pins or fins. Two forms of indirect radiation are illustrated in Figs. 19*a* and 19*b*. The sections are joined together in the same manner as are the sections of direct radiators. The form shown in Fig. 19*b* is of the so-called short-pin type. A similar form having longer pins can also be obtained.

54. Arrangement of Indirect Radiators.—Two common arrangements for indirect radiators taking air from outside are illustrated in Fig. 20 and Fig. 21. The radiator is placed in a chamber or box usually situated in the basement of the building, as close as possible to the base of the flue leading to the room to be heated. The air is admitted to the radiator chamber by a duct or flue from

an opening in the outside wall or from the room above. This duct should be provided with a suitable damper, arranged if

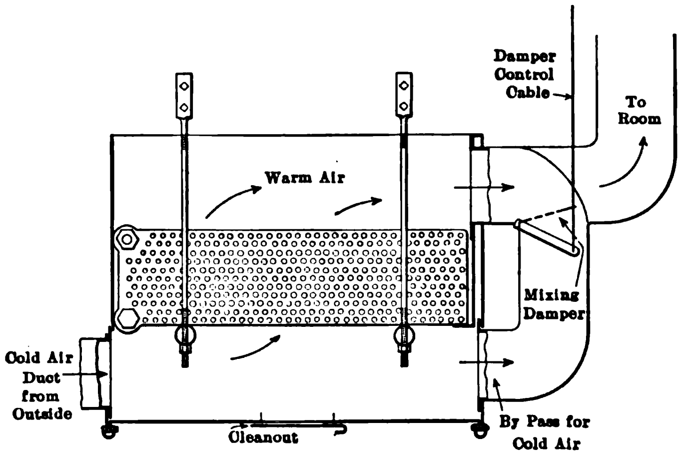


Fig. 20.—Indirect radiator with bypass.

possible to close when the steam or water supply to the radiator is shut off. A bypass damper should also be provided, with a means of controlling it from the room, so that the temperature of the air can be readily adjusted.

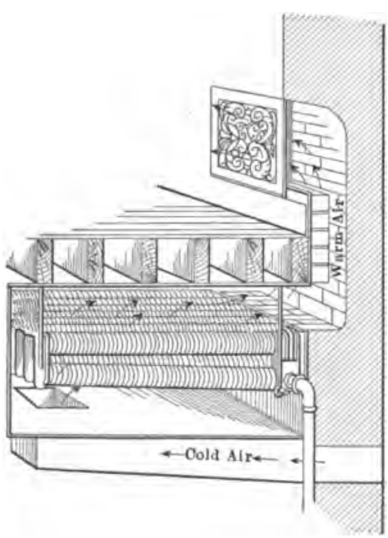


Fig. 21.—Indirect radiator.[1]

The casing surrounding indirect radiators is usually built of galvanized iron and it should be bolted together with stove bolts, so that the sections can be easily removed. A much better method of construction, though a more expensive one, is to enclose the radiator in a brick chamber of sufficient size to permit access to the radiator.

The duct leading from an indirect radiator should be carried to the room as directly as possible. Long horizontal pipes should be avoided.

[1] From "Pipe-fitting Charts" by W. G. Snow.

The indirect radiators are usually suspended in the box or chamber on iron pipes supported by rods from the joists. There should be at least 10 inches clearance between the radiator and the bottom and top of the casing, but the sides of the casing should fit the radiator as closely as possible, so that all of the air must pass through the radiator. Indirect radiators should be placed at least 2 feet above the water line of the boiler if they are to be operated on a gravity steam system, and should be so arranged that the condension will drain from them by gravity. The tappings of these radiators are the same as for two-pipe direct steam radiators. The following table gives the size of flues required for indirect radiators of various sizes.

TABLE XVIII.—SIZE OF FLUES FOR INDIRECT RADIATORS

Heating surface, square feet	Area of cold-air supply, square inches	Area of hot-air supply, square inches	Size of brick flue for hot air, inches	Size of register, inches
20	30	40	8 × 8	8 × 8
30	45	60	8 × 12	8 × 12
40	60	80	8 × 12	10 × 12
50	75	100	12 × 12	10 × 15
60	90	120	12 × 12	12 × 15
80	120	160	12 × 16	14 × 18
100	150	200	12 × 20	16 × 20
120	180	240	14 × 20	16 × 24
140	210	280	16 × 20	20 × 24

Indirect radiators are sometimes arranged to re-circulate the air from the room instead of drawing in fresh air from outside. No ventilation is obtained by such an arrangement and the only advantage of the indirect radiator so installed is that it is concealed.

55. Heat Transmission from Indirect Radiators.—Heat is transmitted from indirect radiators **almost entirely by convection.** The amount of heat which will be transmitted from a given indirect radiator depends upon the temperature of the entering air, the temperature of the radiator, and the quantity of air passing through the radiator. The last quantity depends in turn upon the relative temperatures of the heated air and the unheated air, and upon the friction in the air ducts. In Fig. 22 let h' be the average vertical distance from the radiator to the

point of delivery to the room. The force effective in producing the flow of air is then

$$p = h' (D_1 - D_2)$$

in which D_1 = density of outside air.
D_2 = density of heated air.

During a state of constant flow the quantity of air passing through the radiator will always be just sufficient so that the friction loss due to the air passing through the system will equal the available head producing flow. Owing to the impossibility of determining in advance the resistance of the duct, because of lack of a standard type of construction, it is very difficult to compute accurately the quantity of air which will pass

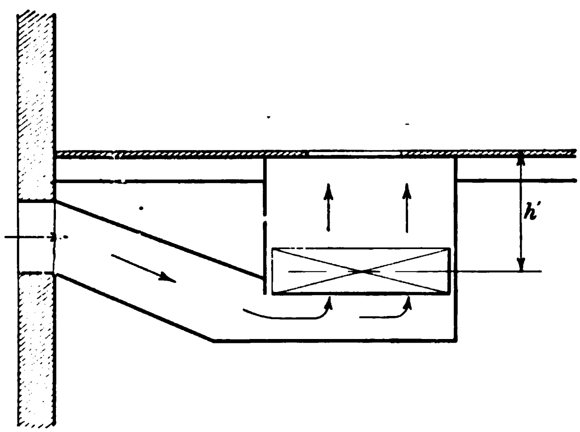


FIG. 22.

through the system. The action is also complicated by the stack effect of the heated room above. Accordingly the methods used in designing indirect radiators are based on experimental data. Table XIX gives the amount of heat transmitted from standard and long-pin radiators under various conditions.

It will be noted that the temperature to which the air is heated by the long-pin radiator is less than that to which it is heated by the short-pin radiator with the same quantity of air passing. This is undoubtedly due to the fact that the pins are so long that the rapid removal of heat by the air causes the ends to become cooled. The long-pin type, however, is very desirable for use when large quantities of air are required, as the air passages are ample. This is the work for which it is primarily designed. The

short-pin type gives better results for ordinary residences and other buildings where only small quantities of air pass through the radiator.

TABLE XIX.—HEAT TRANSMISSION FROM PIN RADIATORS

Cubic feet of air passing per square foot of radiation per hour	Rise in temperature of the air		Pounds of steam condensed per square foot of radiation		B.t.u. transmitted per square foot of radiation per degree difference in temperature between steam and air	
	Standard pin	Long pin	Standard pin	Long pin	Standard pin	Long pin
50	147	140	0.125	0.150	0.80	0.95
75	143	137	0.170	0.210	1.17	1.27
100	140	135	0.240	0.260	1.51	1.60
125	138	132	0.295	0.310	1.85	1.90
150	135	129	0.355	0.360	2.22	2.20
175	132	126	0.410	0.405	2.57	2.47
200	130	123	0.470	0.450	2.90	2.72
225	127	120	0.530	0.490	3.25	3.00
250	123	118	0.585	0.530	3.60	3.20
275	121	115	0.645	0.570	3.90	3.40
300	119	112	0.700	0.610	4.22	3.60

56. Calculation of Indirect Radiation.—In order to determine the required size of an indirect radiator it is necessary to assume the quantity of air that will pass through the radiator. In school buildings and other buildings where a large air supply is desired and where the flues will be of ample size, the amount of air passing per square foot of radiation may be assumed to be 200 cubic feet per hour. In residences and buildings where the flues are usually small, the amount of air passing per square foot of surface per hour does not exceed 150 cubic feet. The air should be assumed to enter the radiator at the minimum outside temperature for which the system is to be designed. If this temperature is 0°, for example, and the quantity of air passing is taken as 200 cubic feet per hour per square foot of radiation, the air will be heated according to figures given in Table XIX to about 130°. The air which enters the room at this temperature gives up its heat to supply the heat lost by conduction through the walls, and finally finds its way out of the room through the window cracks, foul air flues, etc. Each cubic foot of air, therefore, gives up enough heat to lower its temperature from 130° to 70°,

if the latter is the room temperature. This amount of heat is equal to

$$\frac{(130 - 70)}{55} \times 200 = 218 \text{ B.t.u. per square foot of radiator surface.}$$

This amount is available for supplying the heat losses through the walls and the amount of surface in the indirect radiator for the case given above would be equal to the computed heat loss through the walls divided by 218.

57. Approximate Rules for Indirect Heating.—The following approximate rules may be used to compute the amount of indirect heating surface required. This quantity in each case is designated by R.

RULE 1.—For ordinary rooms:

$$R = \left(\frac{\text{wall surface}}{4} + \text{glass surface}\right) \times 0.6$$

For entrance halls:

$$R = \left(\frac{\text{wall surface}}{4} + \text{glass surface}\right) \times 0.75$$

RULE 2.—Figure the heating surface the same as for direct heating and add 40 per cent.

RULE 3.—For rooms on first floor:

$$R = \frac{\text{volume of room, cubic feet}}{40}$$

For second and third floor rooms:

$$R = \frac{\text{volume of room, cubic feet}}{50}$$

For stores and large rooms:

$$R = \frac{\text{volume of room, cubic feet}}{60}$$

58. Combination of Direct and Indirect Radiators.—A very common arrangement is to install enough indirect radiation to give the proper amount of air for ventilation and to install direct radiation to supply the heat losses. The direct radiation would then be computed in the ordinary manner, as if there were no other source of heat. This system has the advantage of being more economical, as less cold air need be heated per hour.

Further, when the rooms are unoccupied, the indirect radiators may be entirely shut off, resulting in a considerable saving of fuel.

59. Semi-direct Radiators.—When only a small quantity of air is needed for ventilation semi-indirect or "flue" radiators may be used in place of indirect radiators. A radiator of this form is shown in Fig. 23. The air enters through a grating in the wall behind the radiator and passes into a metal box which encloses the lower part of the radiator and thence up through the spaces between the sections. Dampers in the fresh air opening and in the base may be adjusted to allow part or all of the air to re-circulate from the room. Radiators used for this purpose are of a special design, the sections being so shaped that the passages between them are divided into a number of vertical flues. A test recently conducted on a flue radiator showed that about 45 per cent. of the ttoal heat transmitted is carried off by the air passing through the flues, the remaining 55 per cent. being given off by radiation and by convection from the outer surfaces. When flue radiators are used the amount of surface allowed should be about 25 per cent. greater than if direct radiation were used.

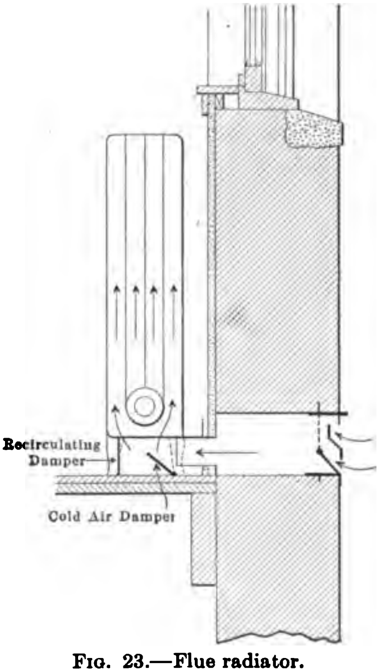


Fig. 23.—Flue radiator.

Problems.

1 To be properly heated, a certain building requires 5627 square feet of 30-inch, one-column radiation. How much would be required if wall coil, of sections containing 9 square feet of surface, long side horizontal, were used? How much would be required if pipe coils, 9 pipes high, were used?

2. A heating system is guaranteed to heat a building to 70° in zero weather at 5 pounds pressure. A test is made with the outside temperature at 10°. What inside temperature must be reached to fulfill the guarantee?

3. A heating system is guaranteed to heat a building to 65° with the outside temperature at 10° and at a steam pressure of 1 pound. A test is made with the outside temperature at 15°. What inside temperature must be maintained to fulfill the guarantee?

4. Assume that the room in Fig. 7, p. 23, is to be heated by indirect radiation. Inside temperature 70°, outside temperature 0°. How much radiation would be required and what would be the proper size for the flues and registers?

5. Take the same room as in Prob. 4 and figure the amount of indirect radiation required by each of the approximate rules in Par. 57.

6. Take the same room as in Prob. 4 and figure the amount of indirect radiation required if the inside temperature is 65° and the outside temperature 10°.

CHAPTER VI

STEAM BOILERS

60. Fuel.—Before taking up the subject of boilers, it is desirable to study the various kinds of fuel and the general principles of combustion.

Coal, coke, wood, oil, and gas are used as boiler fuels. Coal is by far the most widely used fuel in the United States, being found in varying amounts in no less than thirty States in the Union. It is of vegetable origin, being the remains of vegetation which existed during a former geological period and which gradually reached its present state through the action of decay and of earth pressure. The chief constituents of coal are carbon, hydrogen, oxygen and nitrogen. The carbon exists partly in an uncombined or "fixed" state and partly in combination with the hydrogen and oxygen as hydrocarbon compounds which are given off as gases when the coal is heated. Coals are classified as anthracite, bituminous, etc., according to the relative proportions of fixed carbon and volatile matter as given in Table XX.

TABLE XX.—CLASSIFICATION OF COALS

Kind of coal	Composition per pound of combustible		Calorific value per pound of combustible
	Volatile matter	Fixed carbon	
Anthracite	3.0– 7.5	97.0–92.5	14,900–15,300
Semi-anthracite	7.5–12.5	92.5–87.5	15,300–15,600
Semi-bituminous	12.5–25.0	87.5–75.0	15,600–15,900
Bituminous—Eastern	25.0–40.0	75.0–60.0	15,800–14,800
Bituminous—Western	35.0–40.0	65.0–50.0	15,200–13,700

All coals contain more or less non-combustible matter, consisting principally of moisture and ash. The nitrogen in the coal is also a non-combustible but it is customary to treat it as combustible matter. The moisture content of different coals varies from 2 per cent. to as much as 20 per cent. and the ash content from 4 to 20 per cent. by weight of the coal as mined.

It will be noted that the percentages in Table XX are based on 1 pound of *combustible*.

The bituminous and semi-bituminous coals are the most abundant and are the kinds used for most industrial purposes. Many bituminous coals are of the variety known as "caking" coals because, when heated, the lumps fuse together into a solid crust, while the so-called "non-caking" or free-burning coals do not possess this quality. Bituminous coals burn with a characteristic yellow flame and emit smoke unless burned under favorable conditions. They are sold in the sizes given in Table XXI and as "run-of-mine" or ungraded.

Table XXI.—Commercial Sizes of Bituminous Coal

Kind of coal	Will pass through bars spaced	Will not pass through bars spaced
Lump	....	1¼ inches
Nut	1¼ inches	¾ inch
Slack	¾ inch	

The slack coal does not command as high a price as the larger sizes because of its higher ash content and the difficulty of burning it.

Anthracite or hard coal is principally used for domestic purposes and for other conditions where a smokeless coal is required. It ignites slowly but burns steadily with a short blue flame. It is of relatively great density and does not crumble easily. It is marketed in the sizes given in Table XXII.

Table XXII.—Commercial Sizes of Anthracite Coal

Kind of coal	Will pass through	Will not pass through
Rice	¼-in. mesh	⅛-in. mesh
Buckwheat	½-in. mesh	¼-in. mesh
Pea	¾-in. mesh	½-in. mesh
Chestnut	1¼-in. mesh	¾-in. mesh
Stove or range	1¾-in. mesh	1¼-in. mesh
Egg	2½-in. mesh	1¾-in. mesh
Large egg	4-in. mesh	2¾-in. mesh

61. Composition and Analysis of Coal.—Coal consists of carbon, hydrogen, sulphur, oxygen, and nitrogen combined in various

ways, together with moisture and ash. The moisture includes both that originally contained in the coal and that added during storage and shipment. The moisture content of a given coal is determined by subjecting a finely powdered sample to a temperature of about 220°F. for about 1 hour and noting the loss in weight during that time. This method, while not giving an absolutely accurate result, is the one universally employed.

The amount of volatile matter is determined by subjecting a sample of dried coal to a high temperature out of contact with air until there is no further loss of weight, and noting the decrease in weight. The residue left after distilling off the volatile matter consists of the fixed carbon and ash. By burning the sample in an uncovered crucible the fixed carbon can be removed, leaving the ash.

There are two forms of coal analysis—the "Proximate Analysis" and the "Ultimate Analysis." The former consists of a determination of the moisture, volatile matter, fixed carbon, and ash in the manner just described. This is the more useful form of analysis and is the one generally used by engineers. The ultimate analysis, which consists of a determination of the carbon, hydrogen, oxygen, nitrogen, and sulphur, is usually made in a chemical laboratory. In the proximate analysis, the percentages may be reckoned either on a basis of dry coal or coal "as received." In the former case the moisture content is given in addition.

The heat value or calorific value of a fuel is the amount of heat developed by its combustion, expressed in B.t.u. per pound of fuel. The heat value of coal is determined by igniting a sample of known weight in a closed vessel surrounded by water and noting the rise in temperature of the water. From the previously determined thermal capacity of the vessel and water the heat developed can be computed. The calorific value of the various kinds of coal was given in Table XX.

62. Coke.—Coke is the residue left after the volatile matter is driven off from bituminous coal and consists mainly of carbon. It is produced as a byproduct in the manufacture of artificial gas and is also manufactured for various industrial purposes. It is of relatively low density and is consumed rapidly so that when used as a boiler fuel frequent firing is required unless a very deep bed of fire is maintained.

63. Combustion.—Combustion may be defined as the chemical combination of a substance with oxygen which proceeds at such

a rate that a high temperature is produced. Carbon is the principle combustible in coal. When its combustion is complete, it forms carbon dioxide (CO_2); when it is incomplete it forms carbon monoxide (CO). The hydrogen in the coal unites with oxygen to form water vapor and the nitrogen, which is an inert substance, is set free. For economy in fuel consumption it is necessary that combustion be complete and to this end the supply of air must be ample. In order to insure a sufficient supply to all parts of the fuel bed, it is necessary to supply from 150 to 300 per cent. of the theoretical requirements. As all of this excess air leaves the boiler at the flue-gas temperature, it is evident that in the interest of economy the amount of excess air used should be reduced to the minimum required for complete combustion. The best index of the amount of excess air in the percentage of CO_2 in the flue gases. If exactly enough air is supplied the CO_2 content, by volume, of the flue gases would be 21 per cent. In practice, however, the best results are obtained with a CO_2 content of from 10 to 15 per cent., the higher figure being attainable only with mechanical stokers. In the ordinary hand-fired furnaces of heating boilers the CO_2 content of the flue gases ranges between 5 and 13 per cent.

64. Smoke.—Smoke consists principally of unburned carbon in finely divided particles set free by the splitting up of unburned hydrocarbon gases. While the waste represented by the visible products themselves is not great, smoke is an indication of incomplete combustion and consequently of wasted fuel. A great deal of damage is caused annually by smoke and in most communities the making of excessive smoke is prohibited by law.

Smoke may be avoided by the use of anthracite coal, coke, or the semi-bituminous coals, which have little volatile matter, and by insuring complete combustion when coals high in volatile matter are used. When coal containing much volatile matter is placed on a hot bed of fuel, the volatile matter is distilled off. In order that complete combustion of this gas may take place, sufficient air must be supplied and intimately mixed with the combustible gases. Furthermore, the combustion space must be of sufficient size so that combustion can be completed before the gases come into contact with the relatively cold surfaces of the boiler. The air supply must not be so copious or at such a low temperature as to chill the mixture below the temperature required for combustion. These requirements are met by the

use of various appliances and of furnaces of special design which will be discussed later.

65. Comparison of Different Fuels.—It might be reasonably assumed that from the standpoint of economy that coal is the most desirable which has the greatest calorific value per dollar of cost. This is not strictly true, however, as there are other factors which affect the actual economy. Moisture in the coal is undesirable, principally because of the fact that it absorbs heat when the coal is burned and passes up the stack as superheated steam. An excessive amount of ash is objectionable because the cost of its transportation from the mine must be paid and because of the trouble which it causes in the furnace. It obstructs the passage of air through the fuel bed and fuses together into clinkers which must be broken up and removed from the furnace. The formation of clinker is the most troublesome when the ash is fusible at a comparatively low temperature and also is thought to be aided by the presence of sulphur. The latter should therefore not exceed 3½ per cent.

Coals high in volatile matter are undesirable unless the furnace is designed to burn them, for reasons which have been previously stated. For the smaller sizes of coal and for coals which cake heavily a greater draft is necessary and if not available the desired rate of combustion may be impossible of attainment. In general, the smaller sizes of coal cost less per heat unit because of the less demand for them. When purchased in large quantities the price of coal is often based upon the calorific value and ash content. This is a very desirable way to purchase coal.

Where smokeless combustion is desirable or compulsory, anthracite coal is perhaps the most suitable fuel. The facts that it is the cleanest coal to handle and that it requires little attention render it especially desirable for domestic use. Coke is a very good fuel when the firepot of the boiler is of sufficient depth to hold a large quantity of it. Otherwise, a good fire cannot be maintained without more frequent attention than can conveniently be given. Semi-bituminous coals, such as "Pocahontas" and "New River" are capable of being burned in an ordinary furnace with little smoke because of the small amount of volatile matter which they contain.

The bituminous coals contain the greatest heat value per unit of cost, but have some marked disadvantages. Bituminous coal

is particularly dirty to handle, which is a strong argument against its use in residences. It is also difficult to burn it without smoke except in furnaces of special design, intelligently and carefully operated. With the increasing cost of coal and growing scarcity of anthracite, it is becoming more widely used, however, in all classes of work and many special furnaces are being developed for it.

66. Boilers.—Strictly speaking, a boiler is a vessel in which steam is generated by the application of heat. The furnace in which the heat is developed is often practically an integral part of the boiler, however, and the term "boiler" therefore often refers to the combination of boiler and furnace. The primary requirement in a boiler is that it be of sufficient strength to withstand the pressure which is to be carried in it. In boilers used for heating purposes only, this is comparatively simple as the pressure carried rarely exceeds 10 pounds. Secondly, the heating surface must be sufficient in proportion to the grate surface so that the heat will be largely removed from the flue gases before they leave the boiler; and the boiler should be so designed that the flue gases are made to impinge upon and rub along the heating surfaces to the greatest possible extent as this "scrubbing" action increases the rate of heat transfer. The boiler must be so designed that the water may circulate freely to the heating surfaces and the steam pass away from them without restriction. Also, the area of the surface of the water must be sufficient so that the bubbles of steam rising through the water can escape without excessively disturbing the water level. If the liberating surface is restricted or if the steam space is too small, there is a tendency for priming (*i.e.*, the carrying of water into the steam pipes) to take place, particularly when the boiler is being forced. This consideration is more important in a low-pressure boiler than in a high-pressure boiler as the bubbles of steam have a greater volume at the lower pressure. In boilers used for heating purposes, it is desirable to have a large storage of water so that steam will be continuously generated in spite of slight variations in the condition of the fire. A very large volume of water is not desirable, however, when the boiler is operated intermittently as the entire mass of water must be heated whenever the boiler is put into service.

67. Types of Boilers.—The most common type of boiler for heating residences and small buildings is the round cast-iron

boiler shown in Fig. 24. This type of boiler consists of from three to five main castings such as *A*, *B*, and *C* (Fig. 24). The castings are joined by the tapered nipples *N*, *N*, and are drawn and held together by vertical bolts. For a boiler of a given diameter, the amount of heating surface can be varied by the size or number of the intermediate sections such as *B* in the figure. It is reasonable to suppose that the "taller" boilers are somewhat the more efficient since the ratio of heating surface to grate area is the greater. Round boilers may be obtained having rated capacities up to about 1600 square feet of radiation.

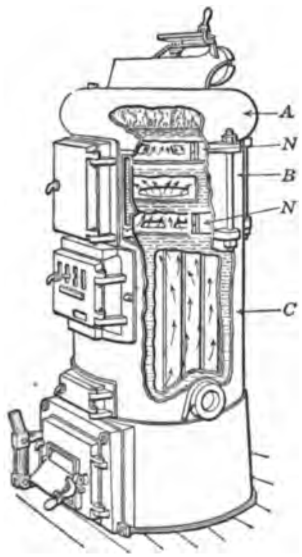


FIG. 24.—Round cast-iron boiler.

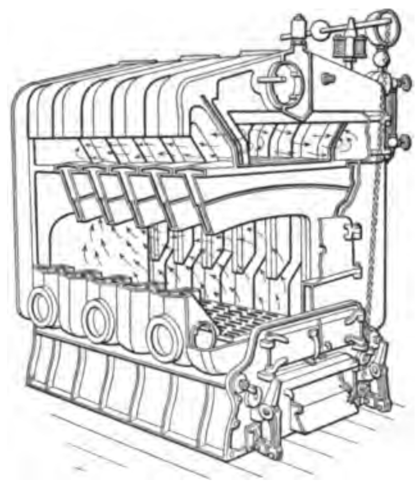


FIG. 25.—Sectional cast-iron boiler.

The "sectional" boiler, as shown in Fig. 25 is obtainable in rated capacities up to about 9000 square feet of radiation. It consists of from five to ten sections joined with nipples. In the larger sizes the sections are made in halves, the idea being to make the boiler capable of being easily transported and erected. One of the advantages of sectional boilers is the possibility of erecting them in an existing building without the necessity of cutting holes in the floor or walls.

Steel boilers are frequently used for heating, particularly in large buildings. A common type is the return-tubular boiler illustrated in Fig. 26. The return-tubular boiler (so named

because the gases flow through the flues toward the front of the boiler) is desirable for heating purposes because of its large water storage, ample circulating areas, and large liberating

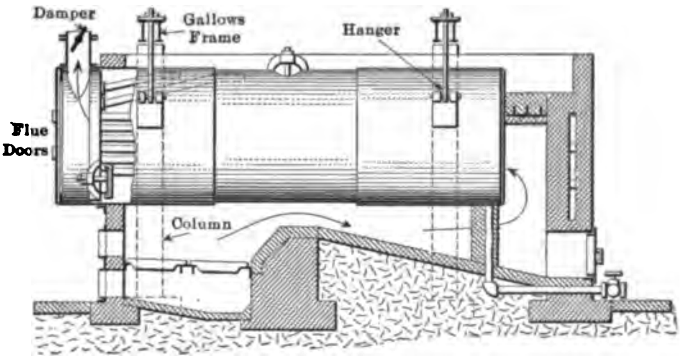


FIG. 26.—Horizontal return-tubular boiler.

surface. Another type of horizontal fire-tube boiler is the firebox boiler shown in Fig. 27. Boilers of this type in which the furnace is incorporated with the boiler are known as portable boilers as

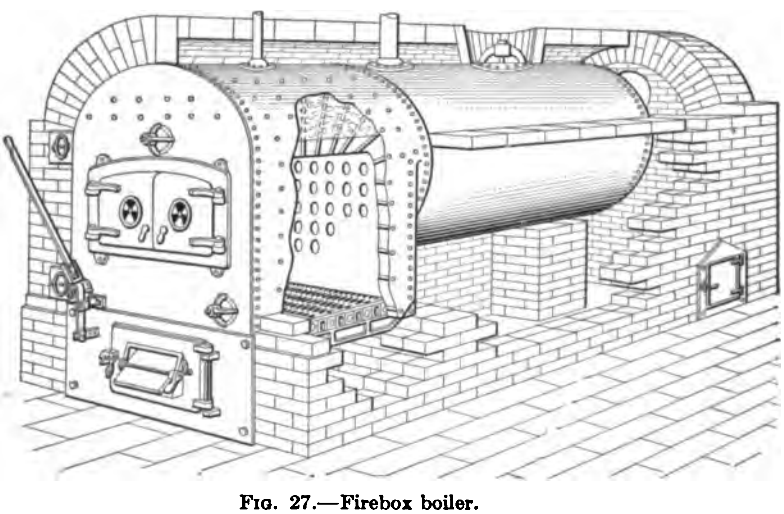


FIG. 27.—Firebox boiler.

distinguished from brick-set boilers of which that in Fig. 26 is an example.

Steel boilers of the return-tubular and firebox types are suitable for working pressures up to 100 pounds. The marine-type boiler shown in Fig. 28 can be used for higher pressures as the fire does not touch the outer shell. Water-tube boilers, in which

the water circulates through the tubes and the flue gases over the outside of them, are used for capacities of over 150 horsepower and for high-pressure work.

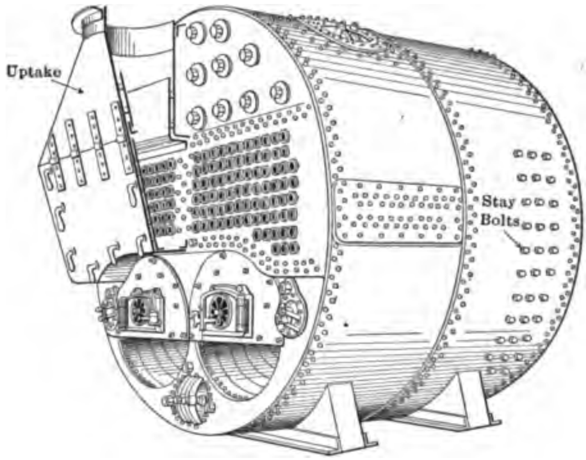


FIG. 28.—Marine-type boiler.

68. Grates.—For heating boilers the grates are usually of the shaking type, consisting of a number of toothed bars as shown in Fig. 29, having a bearing at either end and connected to a rocking link. The free area through the grate is about 50 per cent. of the gross area and the width of the openings varies from $3/16$ to $1/2$ inch, depending upon the size of fuel to be used. In large steel boilers the grates are often stationary and the ashes are removed through the firing door.

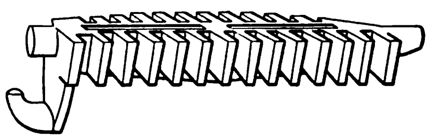


FIG. 29.—Shaking grate bar.

69. The Downdraft Boiler.—Owing to the difficulty of burning bituminous coal without smoke in the ordinary boiler, many boilers have been designed with special furnaces for this purpose, chief among which is the downdraft boiler illustrated in Fig. 30. The furnace consists of two separate grates placed one above the other. Coal is fed to the upper grate only and the air, instead of passing upward through the fuel bed as in the ordinary furnace, enters at the top and passes downward through it. Combustion is most active at the bottom of the fuel bed, and to prevent it from being burned out, the grate is made of hollow bars

through which the water in the boiler circulates. The volatile matter is freed from the coal on the top of the fuel bed and passes down through the incandescent fuel where most of it is ignited. The lower grate contains an incandescent fuel bed consisting of small pieces of coke from which the gases have been driven and which have fallen down through the bars of the upper grate. In the hot combustion chamber between the grates the gases descending from the upper fuel bed mingle with the hot air which enters through the lower grate and complete and smokeless combustion takes place.

In addition to the important feature of burning any grade of coal without smoke and with complete combustion of the volatile

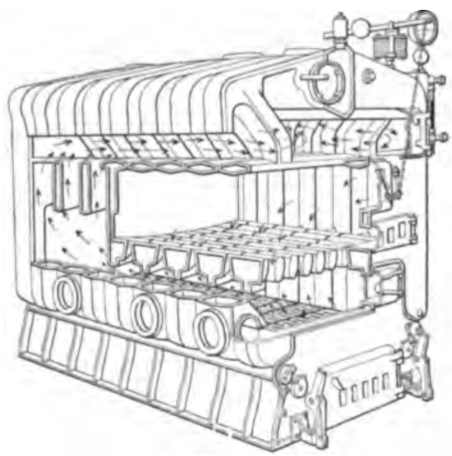


FIG. 30.—Sectional downdraft boiler.

matter, the downdraft furnace has other advantages. No trouble is experienced from clinkers, if the boiler is properly fired, and the performance is uniform as there are no cleaning periods to disturb the fuel bed.

In firing a downdraft furnace, it is important that the main fuel bed be not seriously disturbed. It should be frequently sliced, but just sufficiently to crack the caked mass of fuel so that air can find its way through it. No green coal should ever be fed to the lower grate; it should contain only such material as falls through from the upper grate. The main air supply of course enters through the firing door of the upper grate and the fire is controlled by the regulation of this air opening. The one

great disadvantage of the downdraft furnace is the necessity for fairly careful firing, without which the smokeless feature is lost. If green coal is shovelled on the lower grate, if the lower grate is not properly covered, or if the upper fuel bed is violently disturbed by poking, much smoke will be formed. Any of these things are very liable to be done by a careless attendant.

70. Other Special Furnaces.—Another means of promoting the thorough mixing and combustion of the air and volatile matter necessary for smokelessness is by the use of some form of brick ignition arch or wall. In the boiler shown in Fig. 31 the gases are made to pass from the fuel bed into the "mixing"

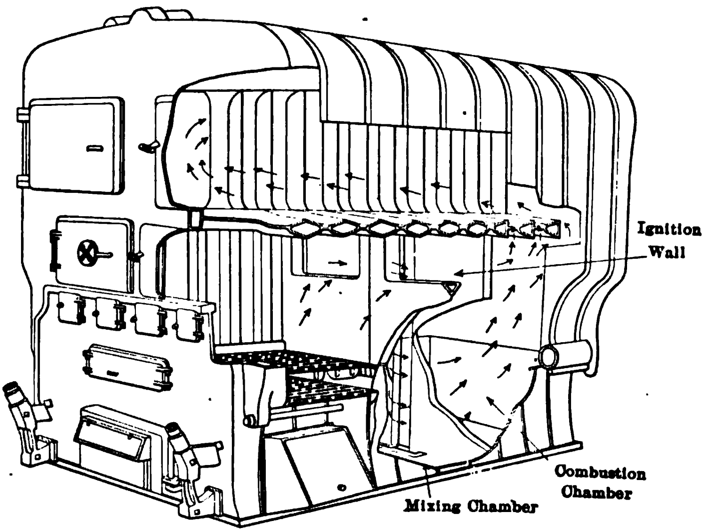


FIG. 31.—Smokeless boiler with brick ignition wall.

chamber and thence through the vertical slot in the ignition wall to the combustion chamber. The ignition wall becomes highly heated and serves to assist in the ignition of the gases, the narrow slot causing a thorough intermingling of the gases and air. The air supply enters principally through the fuel bed and an auxiliary air supply is provided above the fuel bed.

With a boiler of this type, some smoke is unavoidable during the firing periods when the doors are open, admitting great volumes of cold air and when the green coal thrown upon the fire is giving off a large amount of hydrocarbon gases. For the greater part of the time, however, smokeless combustion is obtained.

Other devices for the prevention of smoke consist of ignition arches of various designs, and of steam jets directed into the furnace so as to cause a thorough mixing of the air and gases.

An interesting type of special boiler which is coming into wider use is the magazine-feed type designed primarily for burning the small sizes of anthracite coal and coke. These fuels cannot be burned successfully in an ordinary boiler because of the difficulty of getting air through a fuel bed of any considerable thickness, while a thin fuel bed requires very frequent firing.

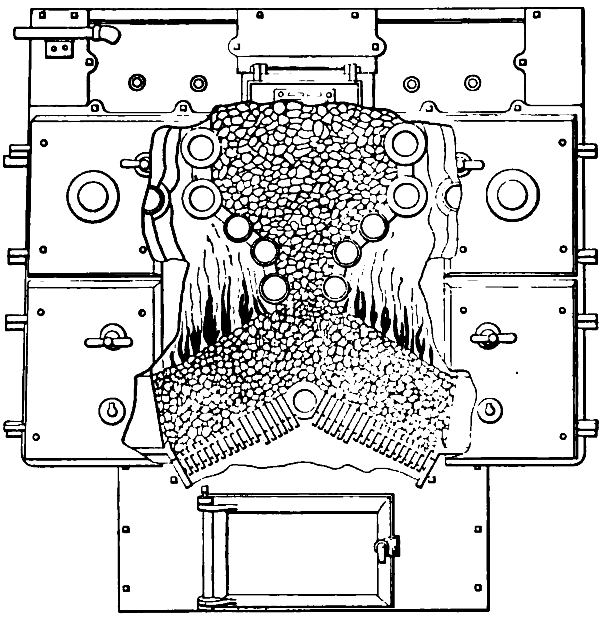


FIG. 32.—Magazine feed boiler.

With the magazine feed such as illustrated in Fig. 32 the fresh fuel is fed by gravity as required and the fuel bed is at all times sufficiently thin to allow air to pass through it. The magazine holds sufficient fuel so that the boiler needs attention only at much less frequent intervals than does the ordinary boiler.

71. Proportions of Boilers.—The heating surfaces in a boiler are defined as those surfaces which have water on one side and hot gases on the other side. In order that the boiler may be efficient the ratio of heating surface to grate surface should be large. The ratio is limited, however, by such factors as the cost

of the boiler and the friction introduced in the path of the flue gases. In small boilers it is usual to allow 1 square foot of grate surface to every 15 to 30 square feet of heating surface. For boilers of 50 horsepower and over, it is usual to allow from 30 to 40 square feet of heating surface per square foot of grate surface, while in very large boilers the ratio is 50 or 60 to 1. Experience has shown that in small heating boilers it is advisable to allow each square foot of heating surface to evaporate only about 2 pounds of water per hour as a greater rate of steaming results in a high exit temperature of the flue gases. In large boilers the evaporation rate varies from 3 to 6 pounds per square foot of surface.

Small heating boilers are distinctly different in operation from large power or heating boilers. In the latter, coal is being fed to the boiler almost continuously and the flues are carrying a large quantity of gases. Small heating boilers, on the other hand, are fed with coal only at infrequent intervals and very little of the heat is transmitted to the water by the flue surfaces, the greater part of the heat being transmitted by the fire surfaces, *i.e.*, those which are in the paths of the heat rays emanating from the fuel bed. During the periods when the drafts are closed most of the steaming in the boiler is produced by the fire surface. It is good practice to have about 60 per cent. fire surface and 40 per cent. flue surface in small cast-iron boilers.

72. Boiler Rating.—The standard unit of boiler capacity is the boiler horsepower which is defined as the equivalent of 34.5 pounds of steam evaporated "from and at" 212° (*i.e.*, from water at 212° into saturated steam at the same temperature). As each pound of steam so evaporated requires the transmission of 970.4 B.t.u., the boiler horsepower is equivalent to 33,479 B.t.u. per hour. It is customary to allow 10 square feet of heating surface per boiler horsepower for establishing the rated capacity of a boiler. Most types of boilers have an overload capacity of from 50 to 100 per cent.

Heating boilers are not usually rated by horsepower, but upon the square feet of direct steam radiation which they will handle. It is never desirable to force a heating boiler up to its maximum capacity as this involves the maintaining of a rapid rate of combustion, necessitates frequent firing, and results in uneconomical operation because the exit temperature of the flue gases is high. The rating is based upon the amount of radiation which the boiler will supply when fired at certain

intervals (usually 8 hours) and with the assumption that the charge of fuel will be entirely consumed except for an amount necessary to kindle the fresh charge of coal. The rating of a heating boiler is therefore largely a function of the capacity of the firepot.

In computing the load on a boiler, allowance should be made for the greater condensing power of indirect radiation and for the condensation taking place in the piping. Three square feet of covered pipe should be considered as being equivalent to 1 square foot of uncovered pipe. It is advisable to be rather liberal in choosing the size of boiler and to allow some excess capacity over and above the total capacity actually required.

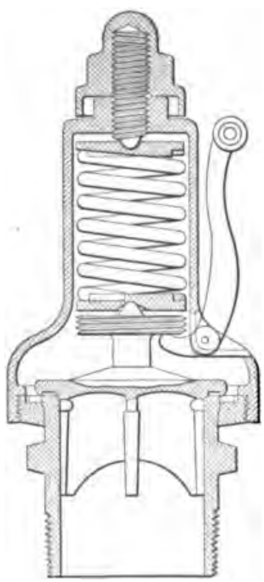


FIG. 33.—Safety valve.

Some engineers install two boilers in buildings of considerable size, each having a capacity sufficient to take care of about two-thirds of the maximum load which could be expected. This practice enables one boiler to be operated at an active rate of combustion during the greater part of the time and provides a spare boiler sufficient to handle almost the entire load if forced. In very large buildings even more spare capacity should be provided.

73. Boiler Accessories.—Every steam boiler should be equipped with a safety valve of sufficient capacity to handle all of the steam which the boiler can generate. A safety valve of the spring-loaded type is shown in Fig. 33. A safety valve of the weight and lever type is undesirable as it can be rendered inoperative through the suspending of extra weights on the lever. The safety valve should be piped a few feet away from the boiler so that a discharge of steam from it will not injure the covering of the boiler. The valve should be set to operate at from 2 to 5 pounds above the normal pressure.

A water column is required to indicate the level of the water in the boiler. It should be equipped with a gage glass and with try-cocks as shown in Fig. 34, the latter being desirable for use in case the gage glass becomes broken or to verify its showing.

A steam pressure gage similar to that in Fig. 35 is also required.

To facilitate the control of the drafts and to maintain an even steam pressure some form of damper regulator operated by the pressure in the boiler is very desirable. The form shown in Fig. 36 consists of a corrugated metal bellows which expands under pressure, closing the ashpit damper and opening the check damper in the flue by means of chains or rods connected to the lever. The pressure at which the action takes

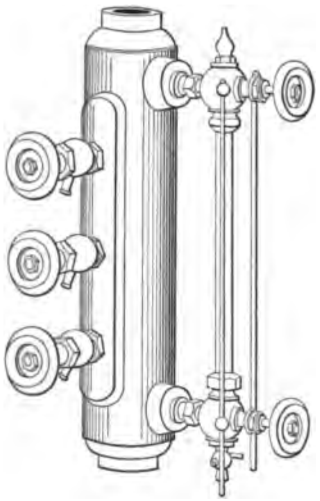


FIG. 34.—Water column.

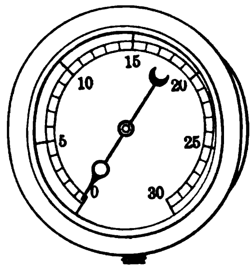


FIG. 35.—Steam pressure gage.

place depends upon the location of the weight on the lever arm.

74. Draft and Chimney Construction.—In order to maintain combustion in a furnace a continuous supply of air must be moved

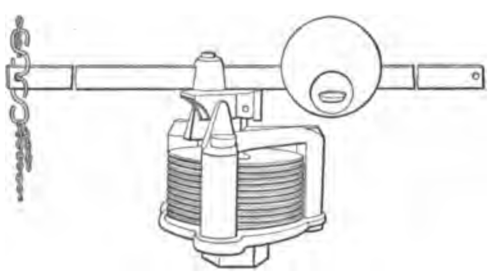


FIG. 36.—Damper regulator.

through the fuel bed. In the ordinary heating boiler, the air is drawn through by means of a chimney, which also serves to dispose of the smoke and other products of combustion. The chimney produces a "draft" or movement of the air because of the difference in weight between the column of hot gases in

the chimney and the cold outside air. The intensity of the force produced depends upon the average difference in temperature between the hot gases in the stack and the outside air and upon the height of the stack. This force must be sufficient to move the required amount of air and gases through the boiler and stack against the frictional resistances interposed by the various obstructions. These resistances consist of (*a*) the resistance of the fuel bed, (*b*) the resistance of the flues of the boiler, (*c*) the resistance of the damper and breeching, and (*d*) the resistance of the stack itself. The first three items are fixed by the kind of fuel used and by the design of the boiler. The last item depends upon the height, cross-section, and construction of the stack. If the cross-sectional area of the stack is too small, the friction in the stack itself will be great and the sum of the various resistance factors may be greater than the available draft produced by the stack. Increasing the area of the stack results in a reduction of its frictional resistance and therefore in an increase in the net amount of draft available at the foot of the stack for overcoming the boiler and breeching losses. Increasing the height of the stack obviously increases the available draft.

TABLE XXIII.—SIZE OF CHIMNEY FLUES

Direct radiation		Height of chimney flue (feet) Diameter of chimney flue (inches)				
Steam in square feet	Water in square feet	30 ft.	40 ft.	50 ft.	60 ft.	80 ft.
250	375	7.0	6.7	6.4	6.2	6.0
500	750	9.2	8.8	8.2	8.0	6.6
750	1,150	10.8	10.2	9.6	9.3	8.8
1,000	1,500	12.0	11.4	10.8	10.5	10.0
1,500	2,250	14.4	13.4	12.8	12.4	11.5
2,000	3,000	16.3	15.2	14.5	14.0	13.2
3,000	4,500	18.5	18.2	17.2	16.6	15.8
4,000	6,000	22.2	20.8	19.6	19.0	17.8
5,000	7,500	24.6	23.0	21.6	21.0	19.4
6,000	9,000	26.8	25.0	23.4	22.8	21.2
7,000	10,500	28.8	27.0	25.5	24.4	23.0
8,000	12,000	30.6	28.6	26.8	26.0	24.2
9,000	13,500	32.4	30.4	28.4	27.4	25.6
10,000	15,000	34.0	32.0	30.0	28.6	27.0

The dimensions of a chimney can be computed from a consideration of the principles stated above,[1] but for ordinary cases

[1] For methods of chimney design see GEBHARDT, "Steam Power Plants."

they can be determined by empirical rules. Table XXIII by Prof. R. C. Carpenter gives the dimensions of chimneys for various amounts of steam or water radiation. If the flue is square, the sides should be equal in length to the diameter for a round flue given in the table, it being assumed that the corners of a square flue are not effective.

The available draft of such chimneys, as measured with an ordinary draft gage, should approximate the values given in Table XXIV.

Table XXIV.—Draft in Small Chimneys[1]

Height in feet	Temperature of chimney gases, deg. F. 200	250	300
	Draft—inches of water		
60	0.27	0.32	0.35
55	0.25	0.29	0.32
50	0.23	0.26	0.29
45	0.21	0.23	0.26
40	0.18	0.21	0.23
35	0.16	0.19	0.20
30	0.14	0.16	0.17
25	0.12	0.14	0.14
20	0.09	0.11	0.12

In measuring the available draft the gage should be connected to the breeching on the chimney side of the damper. The fire should be regulated so that the temperature of the stack gases will approximate working conditions and the damper should be quickly closed immediately before the reading is taken.

A chimney must be so constructed that the wind, deflected by surrounding buildings, will not blow down into it and thus impede the draft. An illustration of two common sources of trouble is given in Fig. 37. The wind striking the sloping roof is deflected over the peak and down into the chimney. The chimney should be extended well above the top of all adjacent buildings.

The round flue is the most effective per square foot of area but is somewhat difficult to construct. For small buildings a square or rectangular flue is used. It should be lined with tile and should be smooth and free from leaks. Offsets should always be avoided, if possible, and when unavoidable should

[1] From "Chimneys: Their Design and Construction," by Harold L. Alt, *Heating & Ventilating Magazine*, March, 1917.

be made with gradual bends. No other openings of any sort should be made in the flue to which the boiler is connected.

In large buildings the stack is often constructed of steel, lined with brick.

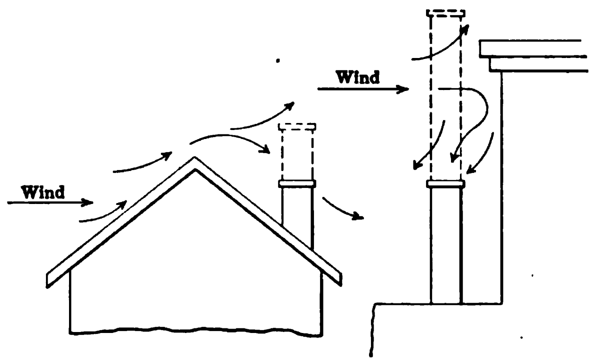


Fig. 37.—Effect of wind on chimneys of insufficient height. Dotted lines show proper construction.

75. Hot-water Heaters.—For hot-water systems the heater used is very similar to the steam boiler. In cast-iron water heaters of both the round and sectional type a smaller casting is substituted for the steam dome. For large buildings ordinary steel boilers are often used, although in many cases the water is heated by the exhaust steam from generating units in some form of "surface" heater.

The water column, safety valve, and pressure gage are of course omitted from a water heater.

PROBLEMS

1. A boiler evaporates 1749 pounds of water per hour from a temperature of 180° into steam at 10 pounds gage pressue and 98 per cent. quality. What is the equivalent evaporation "from and at" 212°, and what boiler horsepower is developed?

2. A boiler containing 820 square feet of heating surface evaporates 2600 pounds of water per hour, from a temperature of 190° into steam at 50 pounds gage pressure and 97 per cent. quality. What per cent. of rating is developed?

CHAPTER VII

STEAM HEATING SYSTEMS

76. Classification of Systems.—In a steam heating system the piping and radiators must be arranged with a view to performing successfully three functions: (1) the conveying of steam to the radiators, (2) the removal of air from the radiators, and (3) the draining off of the condensation from the radiators. The many types of steam heating systems in use differ from one another mainly in the manner in which these operations are accomplished. It is the purpose of this chapter to discuss these various types and their relative merits for different classes of buildings.

Steam heating systems may be divided roughly into two general classes according to the manner in which the connections are made to the radiators. In the **single-pipe** systems the steam is conveyed to the radiator through a pipe which enters the radiator at the bottom of one of the end sections. The condensation which forms in the radiator flows back through this same pipe. In the **two-pipe** systems a separate system of piping is provided to carry away the condensation, and in some cases the air, from the radiators.

77. Single-pipe System.—The simplest form of single-pipe system is that shown in Fig. 38. The nearly horizontal pipes leaving the boiler are called the **steam mains.** The vertical pipes extending to the upper floors are called **risers.** Steam is generated in the boiler and flows through the mains and risers into the radiators, forcing the air out ahead of it through some kind of an air valve on the end of the radiator opposite the supply connection. In the system shown in Fig. 38 the condensation formed in the radiators drains down the risers into the mains and back to the boiler. The direction of the flow of the condensation is thus opposite to the direction of the steam flow. In the risers this is not objectionable if the system is small. In the mains, however, the water and steam flowing in opposite directions are very liable to interfere with each other, unless the mains are of such a diameter that the steam will travel at a very low velocity.

If the pipes are small so that such interference takes place the water is picked up by the steam and driven to the end of the main with a characteristic loud cracking noise known as "water-hammer."

A better design of a single-pipe system is shown in Fig. 39. The main pitches away from the boiler and the condensation

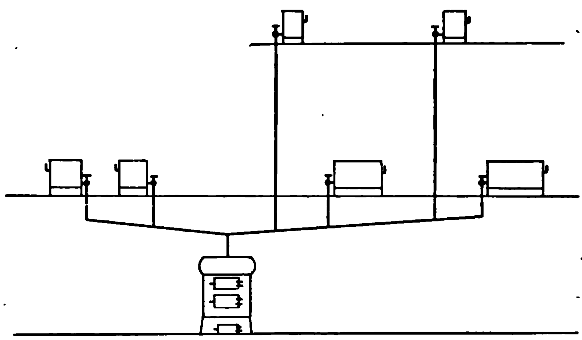


FIG. 38.—Single-pipe system—mains pitching toward boiler.

entering the main from the risers flows along with the steam. The main circles the basement and a drip connection carries the condensation from the end of it to the boiler, entering below the water line. This is the most common form of single-pipe system.

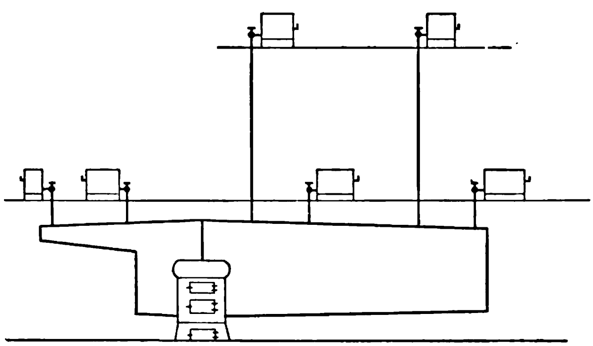


FIG. 39.—Single-pipe system—mains pitching away from boiler.

Another form of single-pipe system is the **single-pipe relief** system shown in Fig. 40. The connections to the risers are taken from the bottom of the main and a drip connection is taken from the foot of each riser to a "wet" return main, so called because it is below the water line of the boiler. The

advantage of this method is that no condensation from the radiators is carried by the main. It also has the advantage of allowing the main to be placed close to the basement ceiling, which is desirable if the basement is used for any purpose for which full head room is desired. This system is sometimes referred to as a two-pipe system because of its return main. It will be noted, however, that there is only one connection to each radiator, as in the other single-pipe systems.

The single-pipe system is simple in design and can be installed at a low cost. It is especially suitable for residences and small buildings where a low-priced system is desired. In large buildings, however, a single-pipe system is less desirable, on account of the large quantities of water which must be carried in the steam

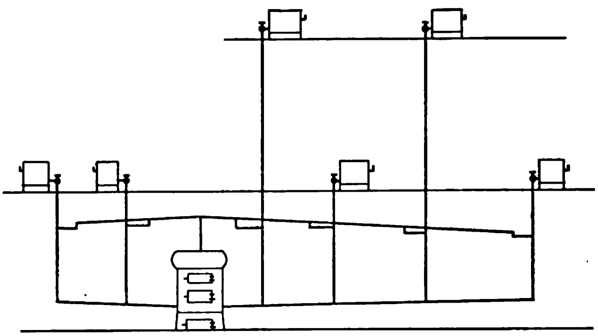


FIG. 40.—Single-pipe relief system.

mains and risers. Another objection is the trouble which is sometimes experienced due to the radiators not draining properly. If the inlet valve is not closed tightly when the radiator is shut off, or if the valve leaks, some steam will continue to flow into the radiator and because of the small area of the opening it is impossible for the condensation to drain out against the inflowing steam. As a result the radiator becomes partly filled with water and when the valve is again opened an annoying cracking and pounding takes place as the water pours out against the inrushing steam.

78. Two-pipe Systems.—Fig. 41 shows a typical two-pipe dry return system. As the term indicates, the return mains are above the water line of the boiler and are filled with steam. The supply mains and risers are installed and connections taken from them to each radiator in much the same manner as in the

single-pipe system. A "return" connection is made from each radiator to the return main, through which the condensation from the radiator flows. As the steam has a free passage through the radiator from the supply main to the return main, it is evident that the latter will be filled with steam at a pressure approaching that in the supply mains, a slight pressure drop taking place through the radiator and its connections. The end of each supply main is dripped into the return main through a 4 or 5-foot seal as at *b*,*b*, which serves to prevent the full steam pressure from entering the return main. One of the chief faults of the two-pipe, dry return system is the tendency for the steam to enter the radiator through the return connection, especially if the

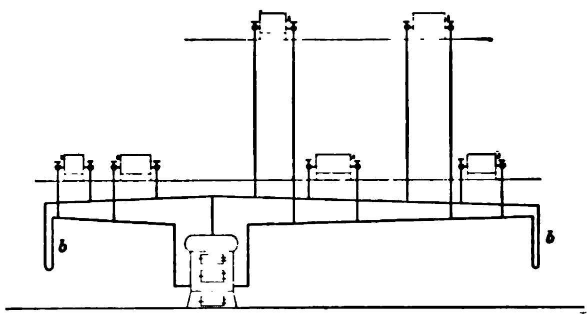


FIG. 41.—Two-pipe dry return system.

return valve is opened first when turning on the radiator, and thus trap air in the center of the radiator.

In the "wet return" system this trouble is eliminated. The return main is below the water line of the boiler and separate connections are made to it from each radiator and from the low points in the supply mains. A wet return system is shown in Fig. 42.

It is evident that no steam can enter the radiator through the return connection, as the lower end of each connection is sealed with water. The water level in the return pipes is sometimes considerably higher than that in the boiler, as will be evident upon consideration of Fig. 42. If the pressure on the surface of the water in the boiler is the same as that on the surface of the water in the return lines, then the water levels will be the same. But if a pressure of 2 pounds, for example, exists in the boiler and there is a drop due to friction, of ½ pound along the main, then the water at (*b*) will rise to a height sufficient to balance the drop

between the boiler and the point (*b*). It is necessary, therefore, to use pipes sufficiently large so that the pressure drop will not be excessive; and futhermore, no radiators should be located less than 2 feet above the water line of the boiler. The wet return system will usually operate with less noise than a dry

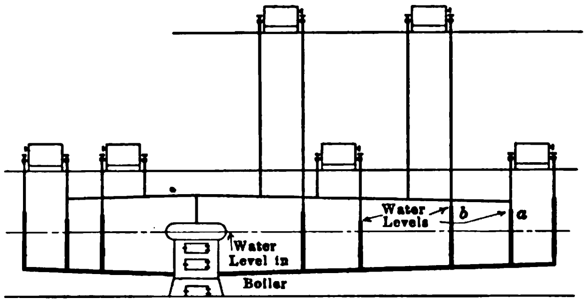


FIG. 42.—Wet return system.

return system as the condensation does not flow in horizontal pipes containing steam. A disadvantage of two-pipe systems is the cost of a double set of radiator valves, and the nuisance of having to operate both valves. Sometimes a check valve is used instead of a shutoff valve on the return end of the radiator.

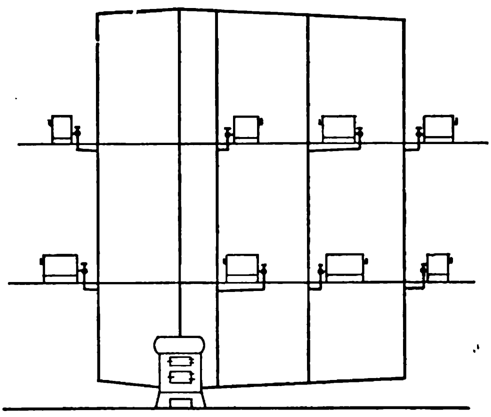


FIG. 43.—Overhead distribution—single-pipe system.

79. Overhead System.—In buildings over three or four stories high the overhead system illustrated in Fig. 43 is nearly always used. The main circles the attic and risers extend down from it to the basement, supplying the radiators on the successive

floors. The steam is carried to the attic main by a main riser from which no radiators are supplied. The chief advantage of the overhead system of distribution lies in the fact that the steam and condensation in the risers are both moving downward. Smaller risers can therefore be used without causing noise or interfering with the circulation of the system. The fact that the large piping is in the attic rather than the basement is also an advantage when the matter of head room and appearance in the basement is a consideration.

The overhead method of distribution may be applied to either the single-pipe or two-pipe system. In the latter case, the return risers and the return main are arranged in the same manner as in the ordinary upfeed system.

80. Air-line Systems.—In the systems previously described, the air is discharged from the radiators through some kind of an air valve to the atmosphere. In order to force the air out of the radiators the steam must be at some pressure above atmosphere, and the temperature of the water in the boiler must be higher than 212°. Consequently, when the fire dies down or is banked at night, no steam is delivered to the radiators. Furthermore, when pressures only slightly above atmosphere exist in the boiler, the radiators near the boiler are wholly or partially filled with steam while those farthest from the boiler may be cold, resulting in an uneven heating of the building. Another objection to the ordinary means of air removal is the disagreeable odor of the air discharged and the noise and frequent leakage of steam and water which are characteristic of most ordinary air valves.

To overcome these objections a system of air lines is sometimes provided to convey the air from all of the radiators to a pump or ejector located in the basement. In place of an ordinary air valve, an "air-line valve" is used, having a pipe connection on the discharge side, and designed to allow air to pass through it but to close against steam. By the suction of the pump or ejector a partial vacuum is maintained in the air-line system and as the steam output of the boiler falls off the vacuum extends into the radiators, piping, and boiler. The boiling temperature is consequently reduced to the temperature corresponding to the existing pressure and the boiler continues to generate steam for a considerable time after the fire is banked. The circulation of the entire system is also improved and a more even heating is secured. In some cases no attempt is made to maintain a vacuum on the

air lines and they are used only to eliminate the ordinary air-valve troubles.

81. Vapor Systems.—A form of two-pipe system having many desirable features is the *vapor* system, which with slight modifications is also variously termed "vacuo-vapor," "atmospheric," etc. These names are derived from the fact that such systems are intended to operate on pressures but little above, and in some cases below atmosphere. The essential features of vapor systems are:

I. *The use of radiators of the hot-water type* with supply valve at the top and with return connection which carries off both the air and condensation.

II. The use of a *graduated supply valve* by means of which the amount of steam admitted to the radiator can be controlled.

III. *Absence of steam in the return lines,* which are either open to the atmosphere or under a pressure less than atmosphere.

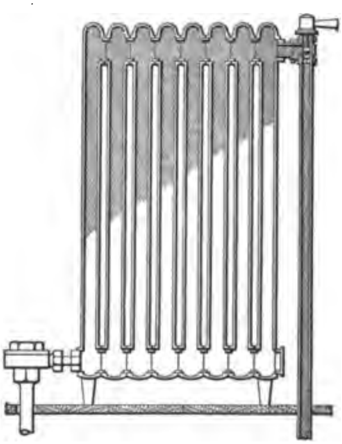


FIG. 44.—Radiator in a vapor system.

The arrangement of a radiator in a vapor system is shown in Fig. 44. By means of a graduated supply valve the steam supply can be controlled so that only the amount required to heat the room is admitted to the radiator. The steam flows into the successive sections of the radiator at the top and fills them through part or all of their length, depending upon the degree of valve opening, in the manner shown in Fig. 44. The surface of the part of the radiator which is filled with steam is at nearly the steam temperature. The remainder of the surface is warmed by the condensation which trickles down the inside surfaces, the temperature decreasing toward the bottom. The temperature of the discharged condensation is thus materially lowered and in cases where the condensation is not returned to the boilers this is an advantage from an economic standpoint.

An important characteristic of vapor systems is that there is normally no steam in the return lines. They carry both the air and condensation from the radiators and are often open to

the atmosphere. The steam is prevented from flowing into the return line from the radiators by either of two means:

(*a*) By some device such as a trap or an orifice installed on the return end of the radiator.

(*b*) By limiting the maximum area of opening of the inlet valve so that at no time will more steam be supplied to the radiator than can be condensed in it.

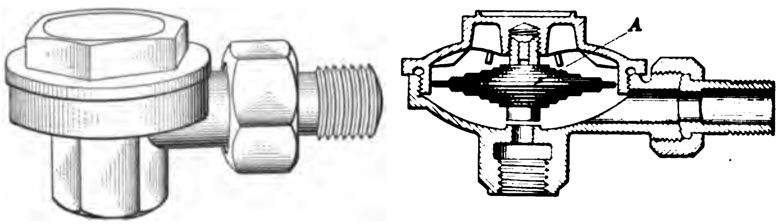


FIG. 45*a*.

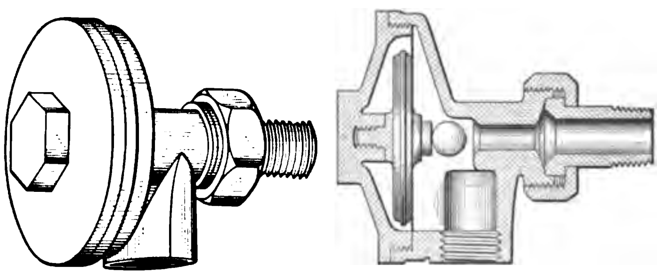


FIG. 45*b*.

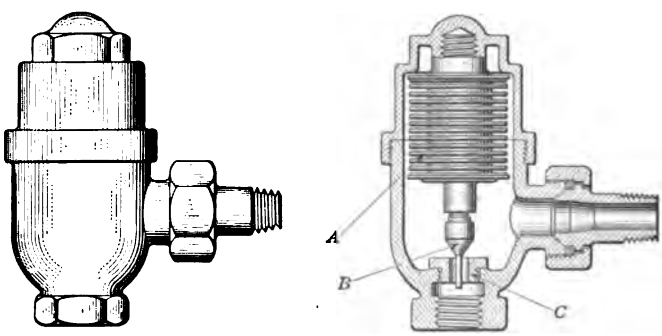


FIG. 45*c*.
Various forms of thermostatic traps.

82. Radiator Traps.—In most vapor systems some kind of a trap is used. The most common is the thermostatic trap which is so constructed as to allow the comparatively cool air and

condensation to pass but to close when the steam at higher temperatures reaches it. Several forms of thermostatic traps are illustrated in Figs. 45*a*, *b*, and *c*. All consist fundamentally of a thin-walled metal chamber *A* (Fig. 45*c*) containing a volatile liquid, such as alcohol, which vaporizes when heated and forms sufficient pressure inside the chamber, at a temperature of about 210°, to expand it and bring the valve *B* against the seat *C*. In operation the trap remains open while air and condensation pass through it but when steam reaches it and heats the thermostatic element it closes, and remains closed until condensation accumulating in it cools a few degrees, causing it to open again and discharge the condensation.

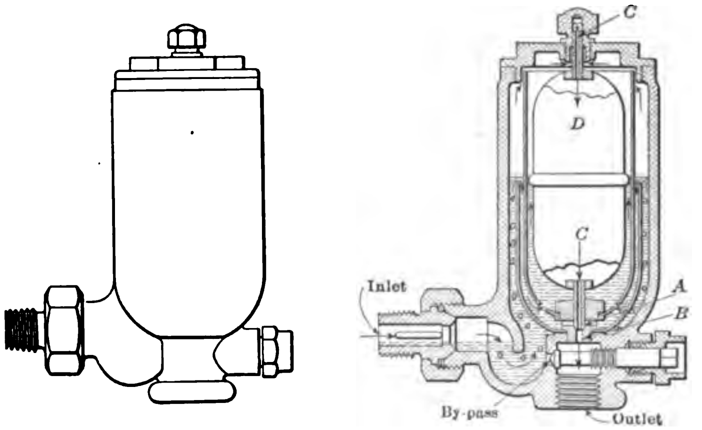


Fig. 46.—Radiator trap of float type.

Another type of radiator trap is the **float** trap in which the opening and closing of the valve is dependent entirely upon the flow of condensation into the trap. A common form is that illustrated in Fig. 46. The valve *A* is normally closed against the seat *B* and the air from the radiator is discharged through the passage *C* in the center of the float. When condensation has accumulated to a sufficient height in the body of the trap, it raises the float *D*, opening the valve and allowing the condensation to flow out until the normal level is reached. The chief objection to float traps is that they are sometimes noisy in operation and are then a source of annoyance to the occupants of the room. Also, there is a tendency for some leakage of steam through the trap to take place.

83. Retarders.—While the thermostatic and float traps are designed to close positively against the steam, another type of return fitting is used which only restricts its passage, allowing a small amount to pass into the return line when the radiator is filled with steam. This is not objectionable as the leakage is usually so slight that it is condensed in the return lines. Retarders are usually in the form of an orifice as in Fig. 47. These fittings have the advantages of being of low cost, of simple construction, and of requiring no adjustment. For systems of moderate size they are quite satisfactory. If, however, the pressure regulation is such that a pressure of over a few ounces may exist in the system there is a possibility of an excessive amount of steam leaking into the return lines, which is very undesirable. Such fittings are often used in connection with a supply valve having a restricted opening such as those used in the atmospheric system described in the next paragraph.

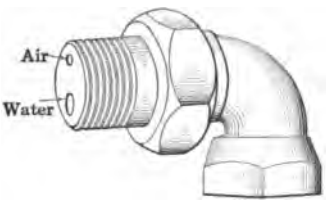


FIG. 47.—Retarder.

84. Atmospheric Systems.—The primary function of the return fittings previously described is to prevent or restrict the leakage of steam into the return line. In the so-called atmospheric system this is accomplished in another way—by restricting the supply so that there will be no uncondensed steam to overflow into the return line. In such systems no special return fitting is provided and the return line is connected direct to the radiator. The maximum area of opening of the supply valve when in its wide open position is restricted by means of an orifice disc, for example, so that with an assumed pressure in the supply pipe—usually about 5 ounces—only the amount of steam which the radiator will condense can enter it. It is evident that the amount of steam which will pass through the maximum opening of the supply valve will vary with the pressure in the supply pipe. Therefore any pressure less than that for which the system is designed will not cause sufficient steam to enter the radiator in the coldest weather. Any considerable increase in pressure above this amount will force more steam through the valve than the radiator will condense and the excess will enter the return piping. If the system has been carefully designed, so that at any one time nearly the same pressure exists at the supply connections of all the

radiators, and if the pressure is closely regulated at the boiler, the atmospheric scheme is quite successful in systems of moderate size.

When the water of condensation is not returned to the boiler, as often happens when steam is obtained from a central heating plant, it is always desirable to utilize the sensible heat in the condensation. Atmospheric systems accomplish this very effectively, the heat being removed as the condensation flows down the walls of a partly filled radiator and through the uncovered return piping. In systems where the steam supply is restricted at the inlet valves the radiators are sometimes given from 10 to 20 per cent. more surface than is required, so that at no time will they be entirely filled and the lower portions are always available for removing the sensible heat of the condensation.

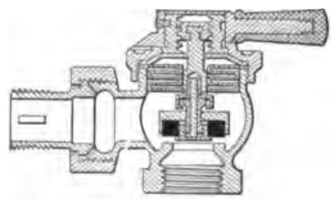


Fig. 48.—Supply valve—maximum opening not restricted.

85. Supply Valves.—The supply valves of vapor systems are of two classes—those which limit and those which do not limit the amount of steam which can enter the radiator when the valve is in the wide open position. In Fig. 48 is shown a valve of the

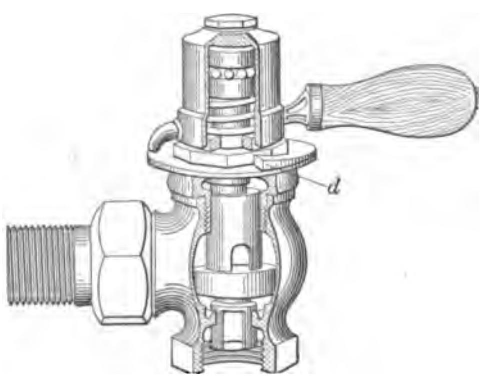


Fig. 49.—Supply valve—maximum opening restricted.

second type. The full opening can be obtained by a half turn of the lever handle and the degree of opening is always readily discernible. The valve can be partly opened according to the amount of heat required. Fig. 49 shows one form of valve

whose maximum opening may be restricted according to the size of the radiator on which it is to be used. The maximum movement of the handle is fixed by the stop (*d*) which is adjusted when the system is first put into service.

86. General Arrangement of Vapor Systems.—The arrangement of the supply and return piping of a vapor system is shown in Fig. 50. The air is forced out of the radiators by the entering steam and passes through the return piping to the air vent located near the boiler. The supply main pitches away from the boiler and is dripped at the end by means of a trap similar to those used on the radiators or by a seal.

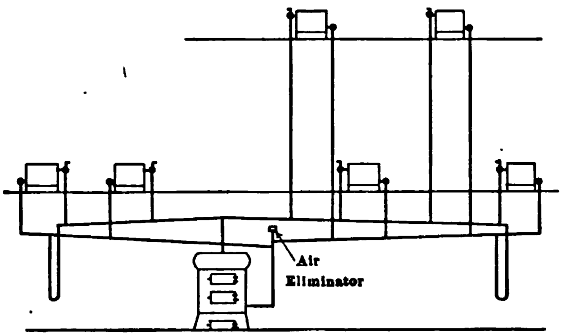


FIG. 50.—Vapor system.

87. Removal of Air from Return Piping.—Many different methods are employed for venting the air from the return piping. The simplest arrangement is to leave the return line open at all times to the atmosphere; but to provide against leakage of steam in case of the failure of a radiator trap to close, a special vent valve is often provided which is normally open and closes only when steam reaches it. These vent valves are quite similar in principle to the ordinary thermostatic radiator trap. Float valves, or combination float and thermostatic valves, are frequently used, their function being to close when water reaches them and thus to guard against leakage in case of the accidental flooding of the return piping.

Some vent valves include also a check-valve arrangement which allows air to escape from the system but prevents it from reëntering. The air is driven out of the system when the radiators and piping fill with steam; and as the steam output of the boiler decreases, the pressure falls below atmosphere and the boiler con-

tinues to generate steam after the temperature of the water in it has dropped below 212°, as is the case in a vacuum system.

88. Advantages of Vapor Systems.—It is apparent that for many classes of buildings vapor systems have some advantages over the other systems of heating, which may be summarized as follows:

1. *Control of the Heat Supply.*—This is accomplished by the manipulation of the supply valves and is therefore dependent for its effectiveness upon the attention of the occupants of the room. The improved design of inlet valve and its accessible location at the top of the radiator render it convenient to operate, but in many classes of buildings the occupants are not inclined to make use of this means of heat control.

2. *Circulation on Very Low Pressures.*—This is of some advantage from the standpoint of economy, but is shared by the various kinds of vacuum systems.

3. *Noiseless Operation.*—As the steam and water flow in separate systems of piping there is no opportunity for water-hammer.

4. *Discharge of Air into the Basement Instead of into the Rooms.*—This eliminates the noise, smell, and drip which accompany the action of the ordinary air valve.

5. *Economy of Operation.*—The opportunity afforded for accurate temperature regulation coupled with the possibility of circulation on very low pressures are productive of some economy. The measure of saving obtained, however, is rather uncertain.

The disadvantages of vapor systems are the cost of the special fittings and appliances and the maintenance of the radiator traps.

89. Vacuum Return Line Systems.—In a "vacuum return line" system radiators of the hot water type may be used, the arrangement being similar to that of a vapor system, or steam radiation can be used with the inlet valve at the bottom. In either case some form of trap is provided on the radiators and a vacuum pump is connected to the return main.

Various kinds of "exhausters" have been devised for use on vacuum return systems but the most satisfactory apparatus is a simple pump. If a high-pressure steam supply is available, a steam-driven pump exhausting into the heating system is the most economical as regards the energy consumed, but motor-driven pumps have the advantage of requiring much less attention and maintenance. A simple plunger pump is shown in Fig. 51. Pumps of this type can be built to operate on steam

pressures as low as 10 pounds but this necessitates a very large steam cylinder. In general, unless steam of at least 25 pounds pressure is available, it is better to use a motor-driven pump.

For the proper operation of a vacuum system it is essential that the traps on the radiators be in good condition and close tightly. If they do not close tightly a leakage of steam into the return pipes will occur which will make it very difficult to maintain the vacuum. A water spray at the vacuum pump suction is often used to condense any steam which may be present, but the use of an excessive amount of spray water is a source of considerable loss, as the spray water must necessarily be wasted, carrying with it the latent heat of the steam which it has condensed.

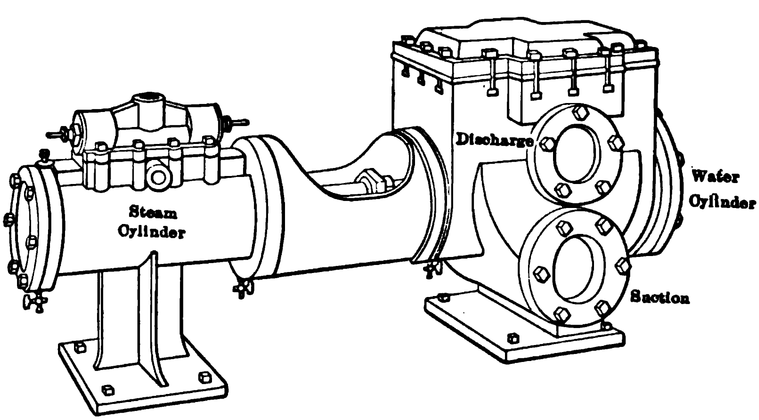


FIG. 51.—Steam-driven vacuum pump.

One of the advantages of vacuum systems—the continued generation of steam at temperatures below 212°—has already been brought out (Par. 80). Another important advantage is the better circulation in both supply and return pipes produced by the greater pressure differential. If, for example, a vacuum system is operated with a steam pressure of 2 pounds and a vacuum of 10 inches of mercury, the total differential is about 7 pounds. A more rapid warming up of the system, better removal of air from the radiators, and better circulation in return lines having air or water pockets are other advantages which might be mentioned. In case some radiators are located, perforce, below the water line of the boiler a vacuum pump must be used to drain them properly. From the standpoint of

economy vacuum systems are of some advantage because of the lower radiator temperatures which exist if a vacuum is carried on the entire system at times when less heat is needed. . When exhaust steam is used for heating a vacuum system permits of a lower back pressure on the engines and turbines and therefore tends to better the economy of the plant. Vacuum systems are best suited to large buildings where the advantages to be gained will justify the initial cost and the operating cost of the special equipment.

CHAPTER VIII

PIPE, FITTINGS, VALVES, AND ACCESSORIES

90. Pipe.—The pipe used for the conveying of steam and water is made of either cast iron, wrought iron, or steel. Because of the low tensile strength of cast iron, pipe made of this material is suitable only for low pres ures, and must have a relatively thick wall. Owing to its ability to withstand corrosion it is especially adaptable for use where it must be buried in soil. Cast-iron pipe is seldom used in heating work.

The pipe ordinarily used in heating and power plants is made from wrought iron or mild steel Steel pipe is much more widely used than wrought iron pipe at the present time being somewhat lower in price and for many purposes equally as desirable as wrought-iron pipe. The pipe commonly furnished to the purchaser under the name of wrought-iron pipe is likely to be steel pipe, so that if wrought-iron pipe is desired it must be clearly specified. It is rather difficult to distinguish between the two materials except by a chemical test. The threads cut upon steel pipe with an ordinary threading die are usually somewhat the more ragged, however, and this affords a rough means of identification. Wrought-iron pipe is believed by many to be more resistant to corrosion than steel pipe, but the degree of superiority in this respect, if both kinds are well made, is problematical.

In the manufacture of wrought pipe the strips of metal, cut to the proper width, are drawn through a bell to the cylindrical form and the edges welded together. In pipe of the smaller diameters a "butt" weld is used and in the larger sizes a "lap" weld.

Wrought-iron and steel pipe are furnished in sizes ranging from $\frac{1}{8}$ inch to 30 inches nominal diameter. In the sizes up to 14 inches the nominal diameters correspond approximately with the inside diameter of the pipe, while in the larger sizes the pipe is designated by its outside diameter. The nominal and actual dimensions of wrought-iron and steel pipe are given in Table

XXVI. Ordinarily it is not desirable to use the 3½, 4½, 7, 9, and 11-inch sizes unless necessary, as these are regarded as odd sizes and their use is being gradually discontinued. For working pressures of over 150 pounds "full-weight" pipe should be specified. Such pipe is selected as being of full card weight per running foot, while ordinary pipe varies somewhat from the standard weight because of slight variations in the thickness of

TABLE XXVI.—STANDARD WROUGHT STEAM, GAS AND WATER PIPE
Table of Standard Dimensions

Diameter			Circumference		Transverse areas		Length of pipe per square foot of external surface, feet	Length of pipe containing 1 cubic foot, feet	Nominal weight per foot, plain ends	Number of threads per inch of screw
Nominal internal, inches	External, inches	Approximate internal diam., inches	External, inches	Internal, inches	External, square inches	Internal, square inches				
⅛	0.405	0.269	1.272	0.845	0.129	0.057	9.431	2,533.775	0.244	27
¼	0.540	0.364	1.696	1.144	0.229	0.104	7.073	1,383.789	0.424	18
⅜	0.675	0.493	2.121	1.549	0.358	0.191	5.658	754.360	0.567	18
½	0.840	0.622	2.639	1.954	0.554	0.304	4.547	473.906	0.850	14
¾	1.050	0.824	3.299	2.589	0.866	0.533	3.637	270.034	1.130	14
1	1.315	1.049	4.131	3.296	1.358	0.864	2.904	166.618	1.678	11½
1¼	1.660	1.380	5.215	4.335	2.164	1.495	2.301	96.275	2.272	11½
1½	1.900	1.610	5.969	5.058	2.835	2.036	2.010	70.733	2.717	11½
2	2.375	2.067	7.461	6.494	4.430	3.355	1.608	42.913	3.652	11½
2½	2.875	2.469	9.032	7.757	6.492	4.788	1.328	30.077	5.793	8
3	3.500	3.068	10.996	9.638	9.621	7.393	1.091	19.479	7.575	8
3½	4.000	3.548	12.566	11.146	12.566	9.886	0.954	14.565	9.109	8
4	4.500	4.026	14.137	12.648	15.904	12.730	0.848	11.312	10.790	8
4½	5.000	4.506	15.708	14.156	19.635	15.947	0.763	9.030	12.538	8
5	5.563	5.047	17.477	15.856	24.306	20.006	0.686	7.198	14.617	8
6	6.625	6.065	20.813	19.054	34.472	28.891	0.576	4.984	18.974	8
7	7.625	7.023	23.955	22.063	45.664	38.738	0.500	3.717	23.544	8
8	8.625	8.071	27.096	25.356	58.426	51.161	0.442	2.815	24.696	8
8	8.625	7.981	27.096	25.073	58.426	50.027	0.442	2.878	28.554	8
9	9.625	8.941	30.238	28.089	72.760	62.786	0.396	2.294	33.907	8
10	10.750	10.192	33.772	32.019	90.763	81.585	0.355	1.765	31.201	8
10	10.750	10.136	33.772	31.843	90.763	80.691	0.355	1.785	34.240	8
10	10.750	10.020	33.772	31.479	90.763	78.855	0.355	1.826	40.483	8
11	11.750	11.000	36.914	34.558	108.434	95.033	0.325	1.515	45.557	8
12	12.750	12.090	40.055	37.982	127.676	114.800	0.299	1.254	43.773	8
12	12.750	12.000	40.055	37.699	127.676	113.097	0.299	1.273	49.562	8
13	14.000	13.250	43.982	41.626	153.938	137.886	0.272	1.044	54.568	8
14	15.000	14.250	47.124	44.768	176.715	159.485	0.254	0.903	58.573	8
15	16.000	15.250	50.265	47.909	201.062	182.654	0.238	0.788	62.579	8

the sheets from which it is made. For extremely high pressures, "extra strong" and "double extra strong" pipe may be obtained. The extra thickness of the walls is added on the inside of the pipe, reducing the internal area and not affecting the outside diameter. These heavier grades are seldom used in heating work.

91. Pipe Threads.—In order that they may be screwed to a tight joint, pipe threads are made with a taper of 1 in 32 with the axis of the pipe, and the threads in the fittings are tapped to the same taper. Pipe threads are commonly made to conform to

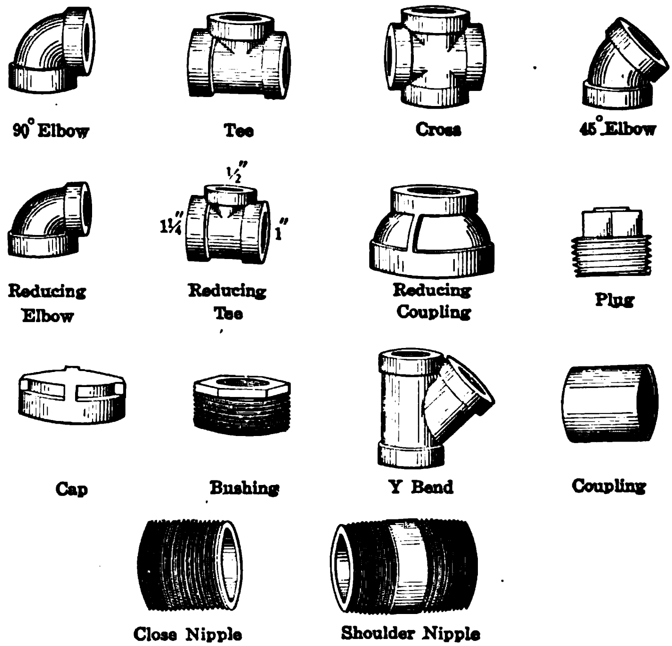


FIG. 52.—Screwed fittings.

the so-called Briggs standard which calls for a thread having a 60-degree angle, with the top and bottom slightly flattened. The number of threads per inch varies for the different sizes of pipe.

92. Screwed Fittings.—The common forms of screwed fittings used in heating work are shown in Fig. 52. All except the ordinary coupling are made of cast iron. In designating reducing tees the size of the openings opposite each other is given first and then the size of the branch opening. For example, the reducing tee in Fig. 52 is a 1¼ by 1 by ½-inch tee.

For pressures over 125 pounds, an "extra heavy" pattern is

available which is suitable for working pressures up to 250 pounds. Extra heavy fittings are made with a greater wall thickness and are of larger dimensions throughout.

93. Unions.—Where screwed fittings are used, provision should be made, at intervals in the line, for disconnecting the piping for repairs, etc. "Right and left" couplings or "unions" are used for this purpose. The former, as the name indicates, are couplings tapped at one end with a left-hand thread, so that both

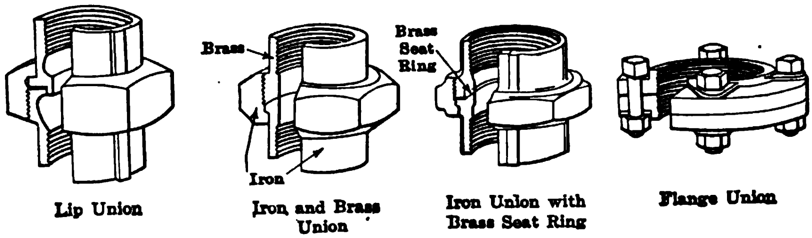


FIG. 53.—Pipe unions.

threads can be screwed up simultaneously. Longitudinal ridges are cast on right and left couplings so that they can be identified after installation.

For pipe sizes up to 2 inches, nut unions, consisting of two pieces screwed to the ends of the pipe and held together by means of a threaded nut are used. Flanged unions are used with larger sizes of pipe. In Fig. 53 are shown these various types of pipe connections. The ground-joint union is superior

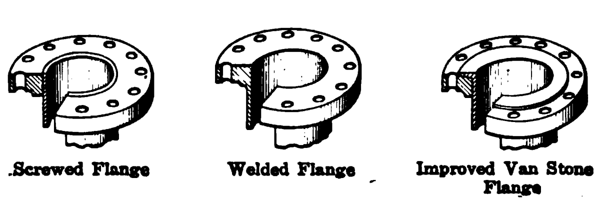


FIG. 54.—Various forms of flanges.

to the gasket union in that it can be disconnected repeatedly without trouble, whereas the gasket in the latter type must be frequently replaced.

94. Flanged Fittings.—Piping of the larger sizes is usually designed with flanged connections, in order that any section of pipe or any fitting can be readily removed. With screwed fittings it is necessary, in order to remove any member, to take

down all of the line from the nearest union or flanged connection. Flanges are commonly screwed to the pipe, especially for low-pressure work. For high-pressure work they may be welded to the pipe or attached by the "Van Stone" method in which the pipe extends through the flange and is formed to a flat face as shown in Fig. 54.

Some forms of standard weight flanged fittings are shown in Fig. 55. These fittings are suitable for pressures up to 125 pounds. There is an extra heavy pattern of flanges and flanged fittings which differ both in general dimensions and in the number and spacing of the bolts.

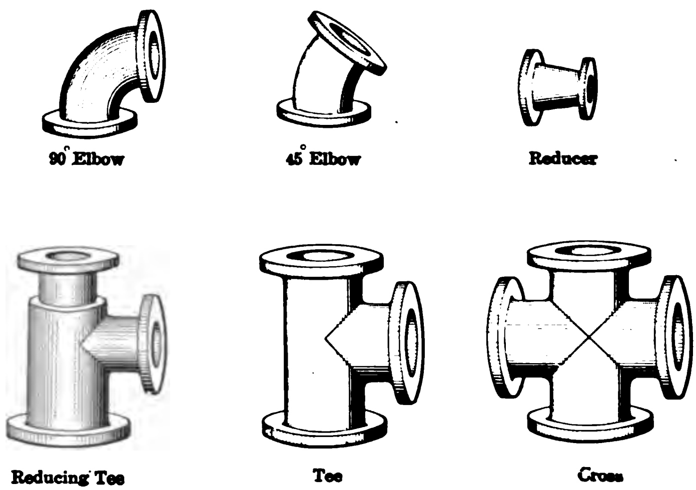
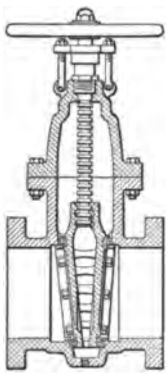


FIG. 55.—Flanged fittings.

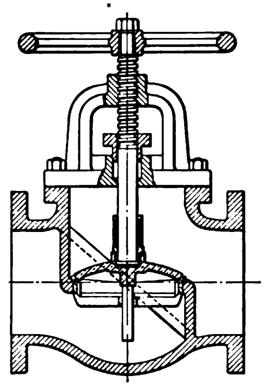
95. Gaskets.—In bolting together flanged fittings it is necessary to insert a gasket between the faces in order to insure a tight joint. Gaskets are made of sheet rubber for water and low-pressure steam lines; for high-pressure lines gaskets of corrugated copper or of various compositions containing asbestos are used. Gaskets are preferably cut in a plain ring to fit inside of the flange bolts.

96. Valves.—In Fig. 56 are shown the various types of valves. The gate valve is the form ordinarily used in steam piping. Globe valves are not permissible in horizontal steam lines as they are so constructed as to dam up the water and cause it to accumulate in the bottom of the pipe, but on vertical steam pipes and on

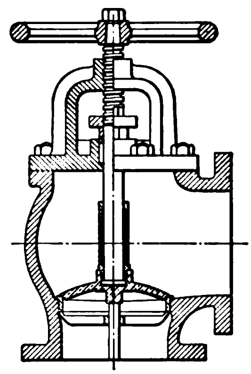
water pipes they are permissible and are especially desirable when the flow of steam or water is to be throttled. The angle valve is a very good type of valve for locations where it can be used.



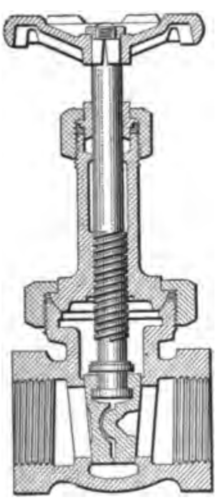
Iron body gate valve non-rising stem.



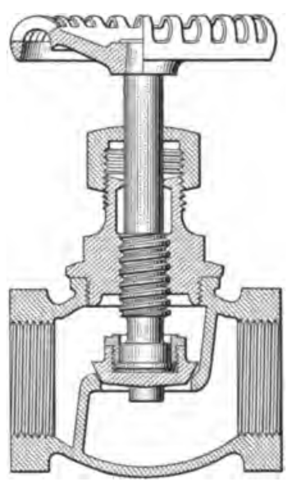
Iron body globe valve rising stem.



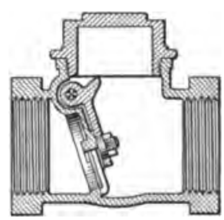
Angle valve.



All brass gate valve.



All brass globe valve.



Swing check valve.

FIG. 56.

Valves in sizes up to 3 inches are made entirely of brass and the larger sizes are usually made of cast iron, with the gates and seats faced with bronze to give a non-corroding surface. The bronze mountings can be replaced when worn. The cover or

"bonnet" of these larger valves is bolted instead of screwed to the body. Gate valves are made either with a "rising" or "non-rising" stem. With a rising stem valve the amount to which the valve is open is always apparent, which is often of great advantage but the space occupied by the valve is somewhat greater.

Check valves are frequently used in heating work. The swing check illustrated in Fig. 56 is the most satisfactory form.

97. Radiator Valves.—The ordinary radiator valve for steam is of the angle pattern and is provided with a union for connecting to the radiator, as shown in Fig. 57. The valve disc is made of

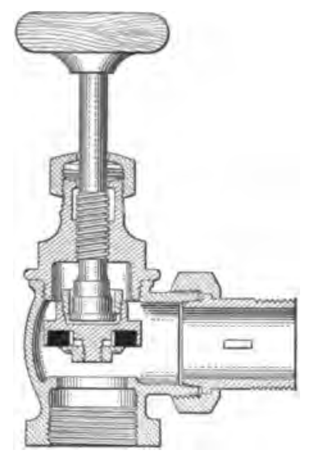


Fig. 57.—Ordinary radiator valve.

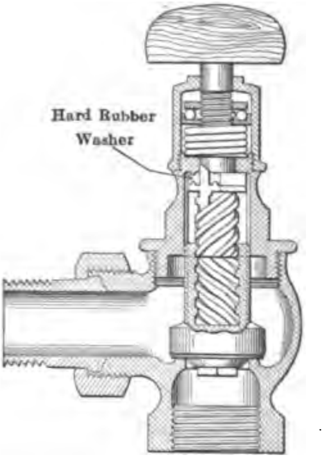


Fig. 58.—Packless valve.

hard rubber and is renewable. These valves are also made in the "corner" pattern.

The stem of the ordinary radiator valve is packed to prevent leakage with a soft stranded packing. The packing is seldom permanently tight, however, and the resulting leakage is often a source of considerable annoyance. In the more modern valves the packing is replaced by a grooved hard-rubber washer which is held against a seat by a spring. The construction of these so-called "packless" valves is shown in Fig. 58. Valves so constructed are much superior to the ordinary type, as all leakage and the necessity of renewing the packing are eliminated.

The ordinary steam-radiator valve may be used in hot-water work. A special hot-water valve is made, however, which

consists of a sleeve having an orifice equal to the pipe area. By a half turn of the hand-wheel the sleeve is turned so that the orifice is brought opposite the opening to the radiator. When closed, the valve allows enough circulation through the radiator to prevent freezing. Fig. 59 shows a valve of this type.

98. Pipe Covering.—The piping of a heating system which is not intended to serve as radiating surface is insulated with some material of low heat conductivity. Most insulating materials owe their useful property to air enclosed in extremely small volumes. If the material is to be an efficient insulator these air volumes must be so minute that the circulation of the air in them is reduced to a minimum and in addition, the material itself must be of low conductivity. A satisfactory pipe covering must also be able to withstand the effect of high temperature and vibration, and to retain its insulating qualities throughout a long period of years.

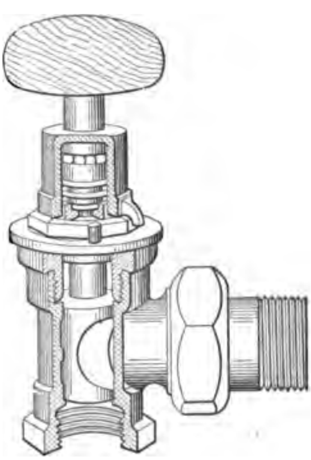


FIG. 59.—Hot water radiator valve.

The material which is probably the most widely used as an insulator is magnesium carbonate. It is in the form of a white powder, and some fibrous material such as asbestos fibers must be used with it as a binder, the aggregate being molded into blocks or into segments curved to fit the various sizes of pipe. Infusorial earth, which consists of the siliceous shells of minute organisms, is also combined with various binding materials to form a very efficient covering.

Many forms of pipe covering are made of asbestos in combination with some cellular material. The compound is rolled into sheets and the covering built up in corrugations so as to enclose air spaces. While not the most efficient type, these coverings are often the most suitable because of their low price. Fig. 60 shows a covering of this type. Hair felt, composed of matted cattle hair, is very efficient but cannot be placed in direct contact with steam pipes owing to its tendency to char at steam temperatures.

In the selection of a pipe covering the cost of the pipe covering

should be balanced against the saving which is effected by the reduction of the heat loss from the piping. The most recent tests made on the commercial grades of pipe covering are those

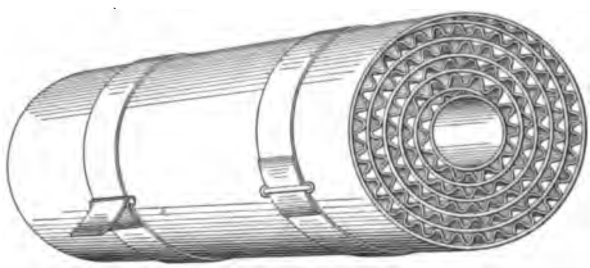


FIG. 60.—Cellular pipe covering.

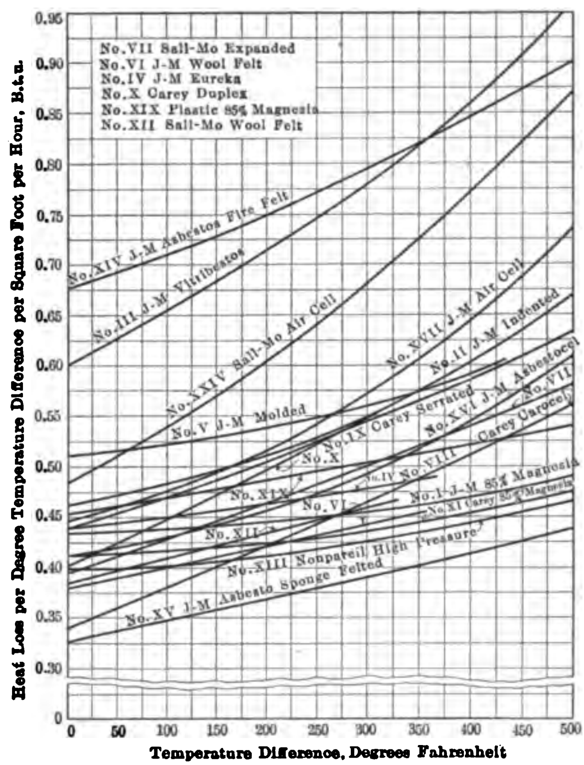


FIG. 61.—Results of tests by L. B. McMillan on single thickness pipe coverings.

of L. B. McMillan and the results of his extensive investigations are shown by the curves in Fig. 61 which give the heat loss

through several commercial coverings of standard thickness for various temperature differences between the surface of the pipe and the air.

It is seldom proper, in heating work, to install the most efficient covering, as the cost of such a covering may easily offset the decrease in heat loss obtained. In fact, the heat radiated from the covered mains and risers of a heating system is not entirely a loss as it is partially utilized. In general, where the steam temperature is high, the service continuous, and the coal expensive a more efficient covering is called for than in the case of low steam pressure and intermittent service, with a low-priced coal.

99. Covering for Boilers and Fittings.—The exposed surfaces of heating boilers are usually covered with an insulating cement, composed of asbestos fibers and various fillers, which becomes plastic when wet. The cement is applied to the hot boiler with the hand to a depth of from 1 to 2 inches and bound with wire, after which a finishing coat of cement and a canvas jacket are applied. The pipe fittings are also covered with cement to the same thickness as that of the pipe covering. For large flanges and fittings removable coverings can be obtained which allow repeated access to the joints without damage to the covering.

100. Air Valves.—In the ordinary steam heating system the air which fills the radiators when they are cold is forced out by the entering steam through some form of air valve installed on the end of the radiator opposite the supply connection. These air valves may be simply hand-operated cocks, which must be opened whenever the radiator is turned on, but the many forms of air valves which allow the air to escape but close automatically when steam reaches them, are greatly to be preferred. Automatic air valves are also designed to close when flooded with water as sometimes happens when a radiator does not drain properly or becomes filled with water because of a leaky inlet valve. The common design is illustrated in Fig. 62*a*. The composition post *A* expands when steam reaches it, causing the valve stem *B* to close against its seat. If water reaches the valve the inverted cup *C*, to which the valve stem *B* is attached, is raised by the buoyancy of the enclosed air and the valve closes. The force thus developed for closing the valve is small, however, and these valves cannot therefore be depended upon to prevent entirely the escape of water. The valve shown in Fig. 62*b* operates on the

same general principle, the expansion of a volatile fluid in the cylinder acting to close the valve when the steam reaches it and the cylinder serving as a float which closes the valve when water reaches it. While more expensive, this form of air valve is more reliable than the cheaper grades. It is always desirable to use air valves of good quality, as the faulty operation of an air valve is a source of extreme annoyance.

Where large quantities of air are to be handled as in the case of a large riser or main, it is better to install a valve with a larger opening than that of the ordinary radiator air valve, so that the air can be discharged in a short time. Such air valves are commonly called "riser vents" and take the form shown in Fig. 62*c*.

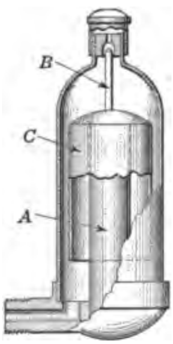


FIG. 62*a*.

FIG. 62*b*.

Air valves.

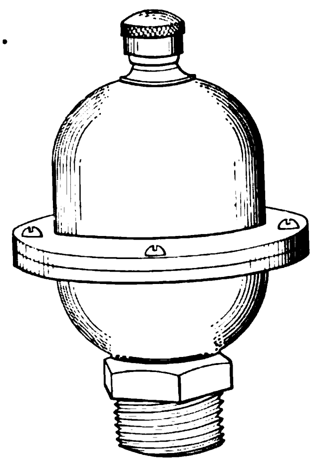


FIG. 62*c*.—Riser vent.

The valves used on an air-line system are intended to close against steam only. If water reaches them it is allowed to run into the air lines, from which it is drained at the lowest point. The expansion member may be either a composition post or a chamber containing a volatile liquid. The latter type is coming into general use. Fig. 63 illustrates these two types.

101. Traps.—A steam trap is a device whose function is to drain the water from a steam pipe, separator, or radiator, without allowing steam to escape. For radiators, special traps of the float or thermostatic form described in Par. 82 are used. For draining steam lines and separators, there are two kinds of traps in use, designated as "float" and "bucket" traps. The former con-

sists of a receiver having a discharge valve controlled by a float in such a way that a raising of the water level from an influx of water causes the float to open the valve, allowing water to be discharged by the pressure of the steam until the water level is

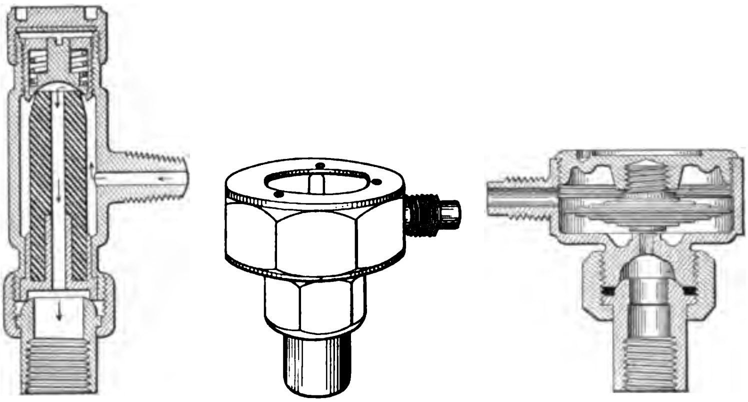


Fig. 63.—Air line valves.

lowered to its normal point. One design of float trap is shown in Fig. 64. A gage glass on the trap indicates the water level. There is normally several inches of water above the valve and the existence of the proper water level affords an indication that the

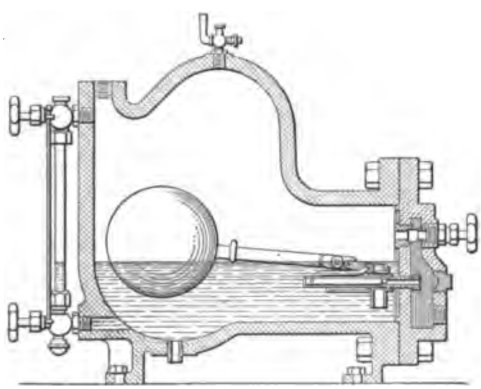


Fig. 64.—Float trap.

trap is operating properly. If the glass is empty, the trap is allowing steam to blow through; if it is full, the trap is not adequately taking care of the water.

The bucket trap consists of a chamber containing a bucket

which is floated by the water in the chamber. To the bucket are attached the valve stem and valve, as shown in Fig. 65. The water flowing into the trap enters and fills the bucket, finally causing it to sink and thereby opening the discharge valve. The steam pressure forces the water out through the valve and empties the bucket, which rises and closes the valve.

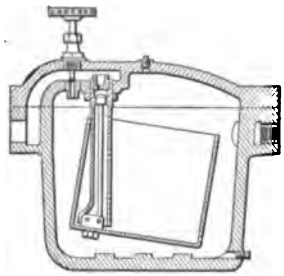


FIG. 65.—Bucket trap.

It is possible to lift the condensation by means of a trap to a height approaching that equivalent to the steam pressure, *i.e.*, about 2.3 feet per pound pressure. It is better, however, if possible, to locate the trap so that it will discharge by gravity.

There is another type of trap which is used where large quantities of water must be handled. This is the tilting trap, one form of which is shown in Fig. 66. The condensation flows by gravity into the chamber which is hinged on the

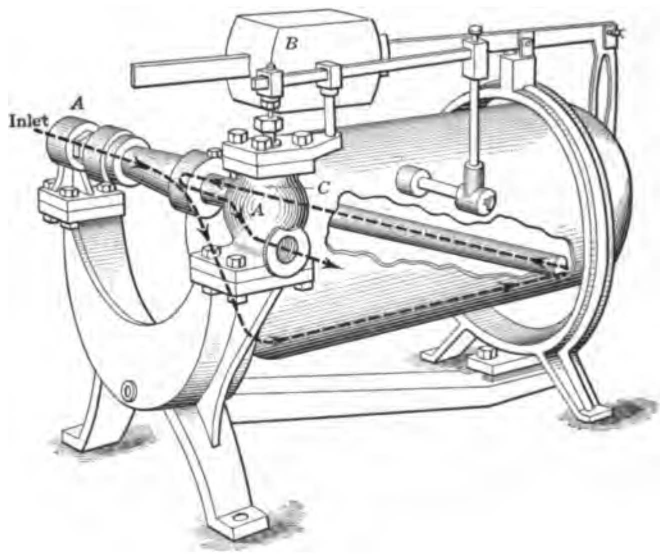
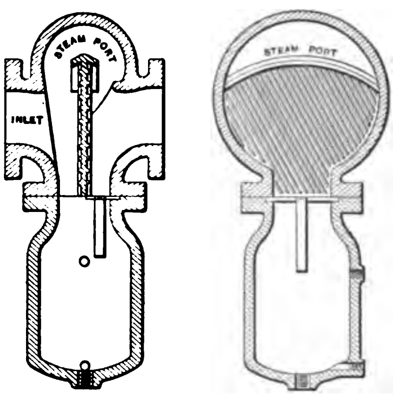


FIG. 66.—Tilting trap.

trunnions A–A and balanced by the weight B. As the chamber fills, the weight B is overbalanced and the chamber falls, opening the discharge valve C. The pressure of the steam forces the water out through the discharge valve and when the cham-

ber becomes empty, it tips back into the filling position and the discharge valve closes. The tilting trap in a slightly different form can be used for lifting the condensation from low-pressure piping to a considerable height, if high-pressure steam is available. In such a trap an additional inlet valve is provided for the high-pressure steam, and the valves are so arranged that when the chamber fills and drops, the main inlet valve closes and the high-pressure inlet valve opens, admitting high-pressure steam which forces out the water and is capable of raising it to any height up to that equivalent to the steam pressure. Tilting traps are sometimes very useful but they require considerable attendance in order to insure their reliable operation.



SIDE SECTION END SECTION

FIG. 67.—Steam separator.

102. Separators.—The function of a steam separator is to remove condensation from steam lines. The separator accomplishes this by abruptly changing the direction of flow of the steam and thereby causing the entrained water to be thrown out, by its momentum, against a suitably designed baffle, usually having a series of grooves which conduct the water into a receiver below. The water is discharged through a trap or seal. This form of separator is illustrated in Fig. 67. Separators are placed in the exhaust line from pumps and reciprocating engines, where they remove the oil as well as the water from the steam. In choosing a separator care must be taken to select a size corresponding to the quantity of steam flowing rather than to the size of the pipe line, for a certain velocity through the separator is necessary to insure the elimination of the water.

103. Reducing Valves.—Steam is occasionally supplied to a building at a pressure much higher than is necessary or desirable for heating purposes, making it necessary to employ a reducing valve, a simple form of which is illustrated in Fig. 68. The pressure on the reduced pressure side of the valve is transmitted through the balance pipe to the under side of the diaphragm, tending to close the valve. The force thus exerted is balanced by

that due to the weights *w–w*, and the valve will assume such a position that just enough steam will pass through it to maintain the required pressure on the reduced side, which pressure is governed by the position of the weights on the lever arm. The reduced pressure may be changed as desired by shifting these

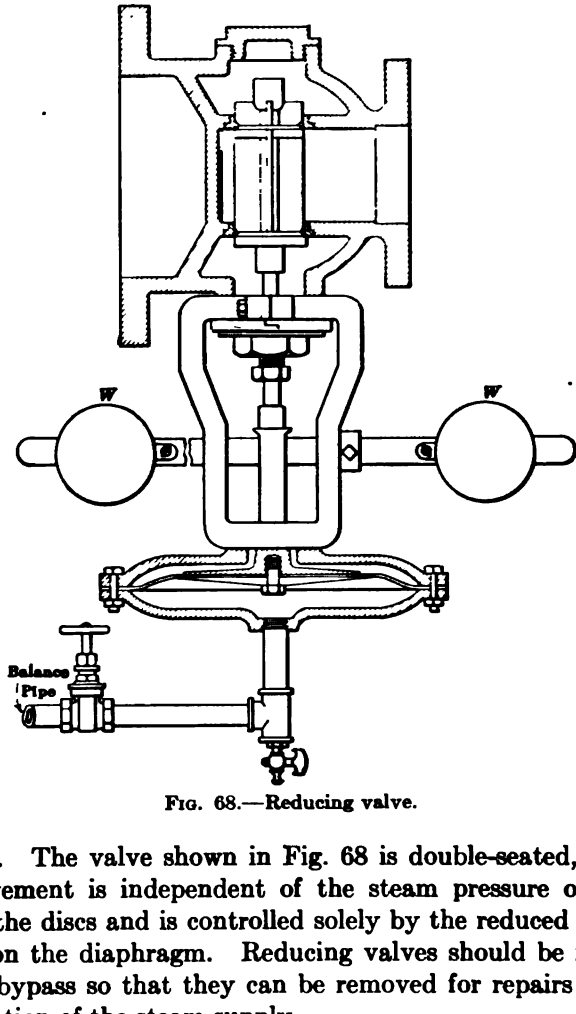


FIG. 68.—Reducing valve.

weights. The valve shown in Fig. 68 is double-seated, so that its movement is independent of the steam pressure on either side of the discs and is controlled solely by the reduced pressure acting on the diaphragm. Reducing valves should be installed with a bypass so that they can be removed for repairs without interruption of the steam supply.

CHAPTER IX

STEAM PIPING

104. General Arrangement.—The elementary arrangement of the different systems of steam heating was shown diagrammatically in Chapter VII. Most of the principles involved in the design of the piping apply equally to all of them.

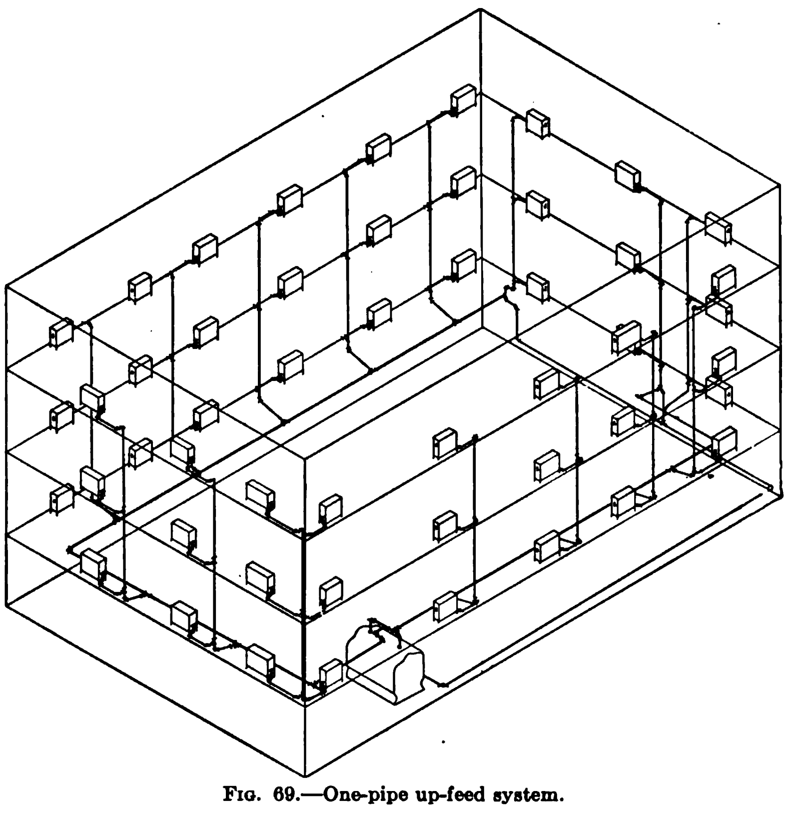


FIG. 69.—One-pipe up-feed system.

In Fig. 69 is shown the piping for a single-pipe upfeed system. The supply mains circle the basement, pitching away from the boiler, and are dripped at the ends into the return main. For

a two-pipe system, the return mains and risers would be arranged in a similar manner.

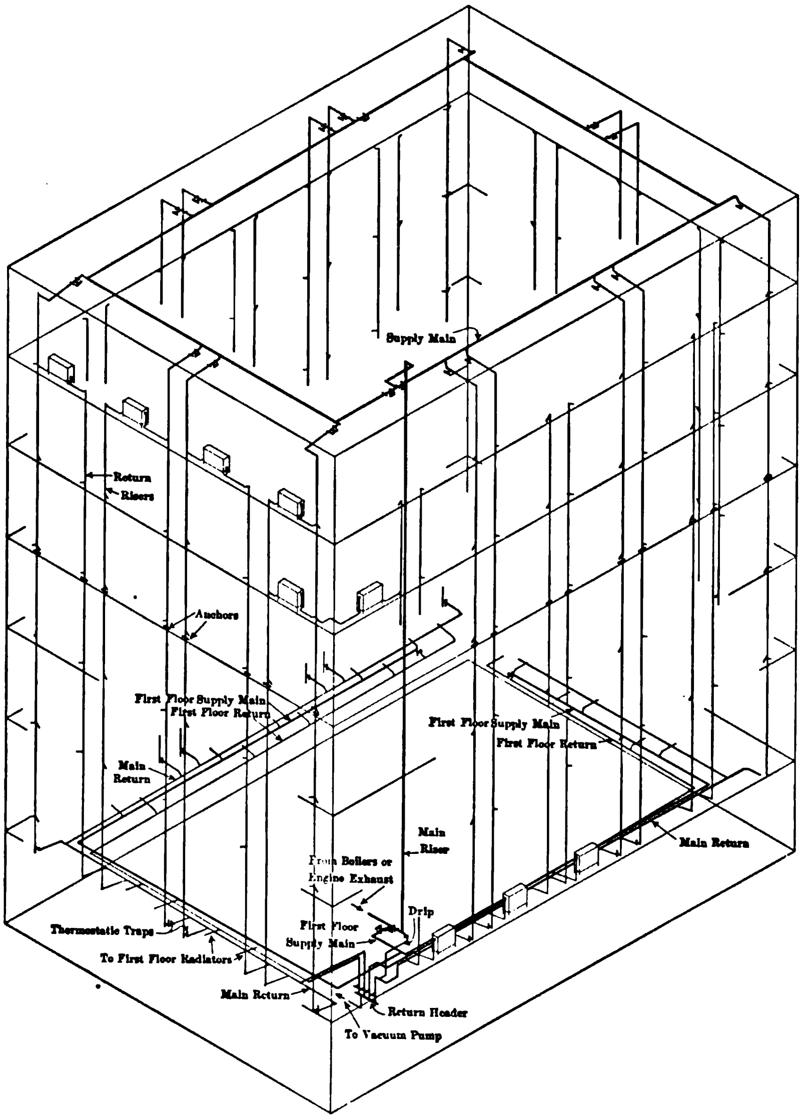


Fig. 70.—Overhead vapor or vacuum system.

Fig. 70 shows an overhead vapor or vacuum system in a tall building. Return risers extend from the top-floor radiators to

the basement, where they tie into the main return line. In large buildings the first floor is often divided into small stores which require heat at times when none is needed in the remainder of the building and *vice versa*, making it desirable to install a separate main to supply the first-floor radiators and arranged so that it can be controlled independently of the main heating system. This scheme also has the advantage of making it much easier to install connections to the first-floor radiators which are often so located that it is difficult to reach them from the risers. In extremely tall buildings it is better to feed the risers from the bottom as well as from the top and a supply main is installed in the basement for that purpose.

105. Principles Involved in Piping Design.—In designing and installing a system of piping, attention must be given to the following fundamental requirements:

1. *Provision for expansion.*

2. *Proper drainage of condensation from the steam lines.*

3. *Proper arrangement of piping and use of pipes of the proper size,* so that the pressure drop due to friction will be small.

106. Expansion.—Perhaps the most important consideration is the proper provision for the linear expansion of the pipes. When steam is turned into or shut off from a system of piping, a change of temperature of the pipe of 140° to 170° takes place and provision must be made for allowing the resulting change of length to occur without putting excessive strains on the fittings. The curve in Fig. 71 shows the theoretical expansion of wrought-iron pipe due to an increase in temperature from 60° to the temperature corresponding to various steam pressures. The temperature of 60° is assumed to be that at which the piping is originally installed. For low-pressure piping a rough rule is to allow 1½ *inches of expansion per* 100 *feet length of pipe.*

The force which an expanding pipe is capable of exerting is extremely great. If constrained at the ends with sufficient rigidity the increase in length will cause the line to "bow" in the center, and the enormous strain thus imposed upon the flanges and fittings is almost certain to crack them. In designing any pipe line some point should be selected as a fixed or anchored point and a comprehensive study made of the amount and direction of the expansion. Provision must be made for properly taking care of the elongation of all parts of the system.

There are in general two ways in which the expansion in a

system of piping may be absorbed: (*a*) *by the turning of some of the threaded joints* and (*b*) *by the bending of the pipes.* The former method is necessary when the expansion is great but small movements can be readily absorbed by the spring of the pipes. Combinations of the two methods are also employed, as will be shown later. For very long and large pipes slip joints or special expansion fittings may be necessary but their use should be avoided wherever possible.

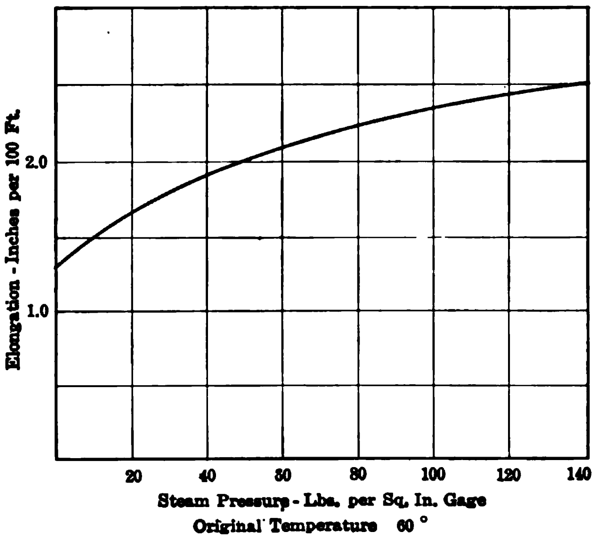


FIG. 71.—Elongation of wrought iron pipe with various steam pressures.

107. Drainage.—There is always some water in pipes carrying saturated steam. In some kinds of heating systems, in addition to the condensation formed in the pipe itself there is also condensation from other pipes and from the radiators. The proper provision for the flow and drainage of the water is important. In horizontal pipes the water should if possible travel in the same direction as the steam and to accomplish this the pipes should be given a pitch of *at least* 1 *inch in* 10 *feet* in the direction of the flow. In case it is necessary to drain the condensation against the flow of the steam, as in a branch to a riser, a much greater pitch should be allowed and pipes of larger diameter should be used so that the velocity of the steam will be low. Any necessary pockets or low points where water might accumulate should be dripped.

108. Mains and Branches.—Horizontal mains are usually anchored at the boiler and allowed to expand freely from that point. The amount of movement of any point along the length of the pipe is evidently proportional to its distance from the fixed point. In connecting risers and branches the movement

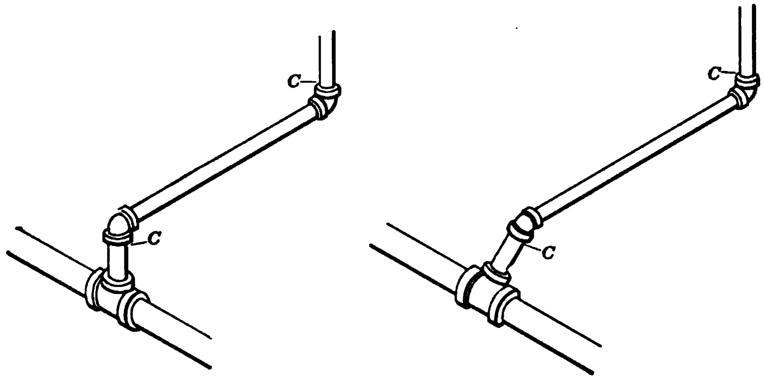


FIG. 72*a*. FIG. 72*b*.
Methods of connecting branches.

of the main is best taken care of by either of the arrangements in Figs. 72*a* and 72*b*. As the main moves longitudinally the threaded joints *c–c* turn slightly. The arrangement of Fig. 72*b* is somewhat the better as the 45-degree elbow offers less resistance to the flow of steam than the 90-degree elbow in Fig. 72*a*. The

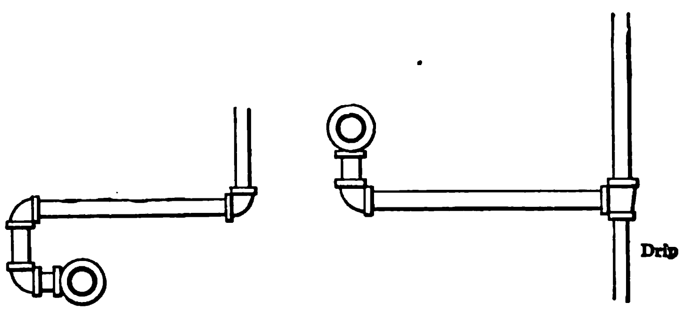


FIG. 73. FIG. 74.

expansion of the horizontal branch is taken care of by the spring of the riser, which arrangement is quite permissible as such branches are rarely over 10 feet long. The arrangement shown in Fig. 73 is sometimes used when the expansion of the main is great. It has the disadvantage of offering considerable resist-

ance to the flow. Branches are sometimes taken from the bottom of the main as in Fig. 74. It is then necessary to install a drip connection in the manner shown. This arrangement is undesirable in one respect. If for any reason the water level rises in the return system above the horizontal connection to the riser, then the riser will be entirely sealed from the main and its steam supply will be cut off. The one-pipe relief system is usually piped in this manner.

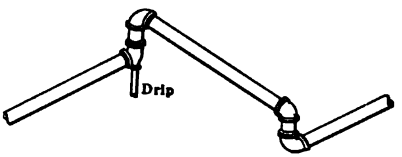


Fig. 75.—Expansion swivel.

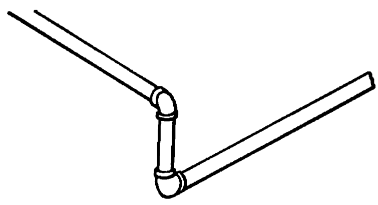


Fig. 76.

In very long horizontal mains in which the movement would be too great to be absorbed by the branch connections it is necessary to anchor the pipe at two or more points and to provide a swivel joint of the form shown in Fig. 75. One objection to this method is the resistance to the flow of steam offered by the fittings.

Another scheme which is sometimes used where the main makes a turn of 90 degrees is that shown in Fig. 76. It will be

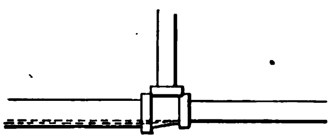


Fig. 77a.

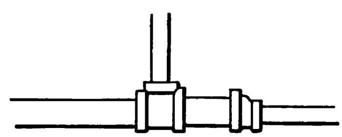


Fig. 77b.

Advantage of eccentric reducer.

noted that this does not give a perfect swivel joint but that the expansion must be partly absorbed by the spring of the members.

When the size of the main is reduced an eccentric reducer should be used as in Fig. 77b so that no water pocket will be formed. The accumulation of water in shallow pockets such as that formed by the reducing tee in Fig. 77a gives rise to severe cracking and pounding when the heating system is started up.

109. Risers.—In small buildings where the supply mains are in the basement, the expansion of the risers is usually downward

and the movement is taken care of by the spring of the branches and by the turning of the tees connecting the branches to the main (see Figs. 72*a* and 72*b*). In larger buildings, where there is a main in the attic, the risers are anchored near the middle and the expansion takes place in both directions. When the expansion is too great to be handled by an ordinary branch connection the

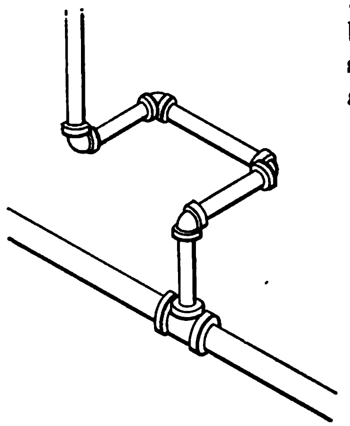


FIG. 78.—Flexible connection for riser.

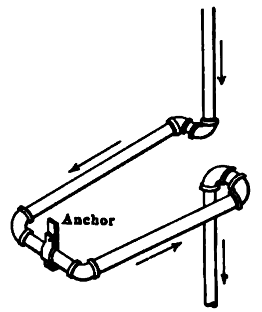


FIG. 79.—Expansion loop for riser.

arrangement in Fig. 78 may be used. This gives a perfect swivel joint and is especially serviceable when the basement main must be installed near the foot of the risers. Its disadvantage is the resistance to the steam flow offered by the fittings.

The branch connection shown in Fig. 72*b* will easily take care of the expansion of risers about four stories high, and that in Fig. 78 about eight stories. For taller buildings an expansion loop of the form shown in Fig. 79 is used. Such an expansion loop is easily capable of handling a length of riser of at least four stories in either direction and gives perfect flexibility. Space is required in the furring to conceal the loop.

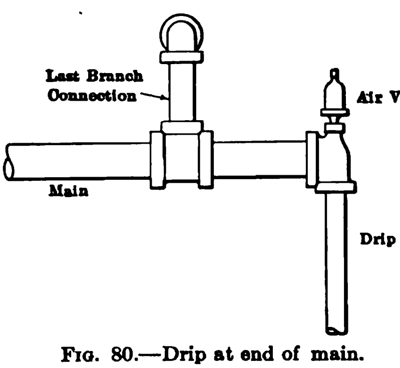


FIG. 80.—Drip at end of main.

110. Drip Connections and Air Venting.—The ends of mains are dripped in the manner shown in Fig. 80. An air valve should

be installed at such points to free the main of air when the system is started up. Drips from different mains should not be connected together above the water line as the pressure of the steam in them may be different, in which case the flow of the condensation would be interfered with and a water-hammer set up.

Air vents should be located at the ends of all mains and at other places where air is liable to become pocketed.

111. Pipe Hangers.—The piping in a heating system must be substantially supported either from the building structure or from special supports. Horizontal mains are usually hung from the joists or steel work of the floor above. For pipes of moderate size the hanger shown in Fig. 81 is widely used. The perforated

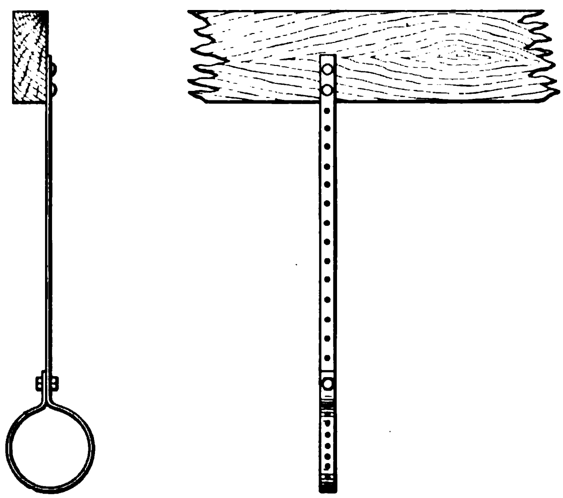


FIG. 81.—Simple form of pipe hanger.

metal can be obtained in long strips and cut to any required length. This hanger is of low cost and can be installed very cheaply.

For heavier pipes the hanger shown in Fig. 82 is a common design. The turnbuckle is used to adjust the elevation of the pipe when it is being installed. Both of these hangers permit the free longitudinal movement of the pipe line. Hangers should be placed at intervals of 20 feet or less on all horizontal pipes.

Risers are supported at the anchor points in some such manner as is illustrated in Fig. 83.

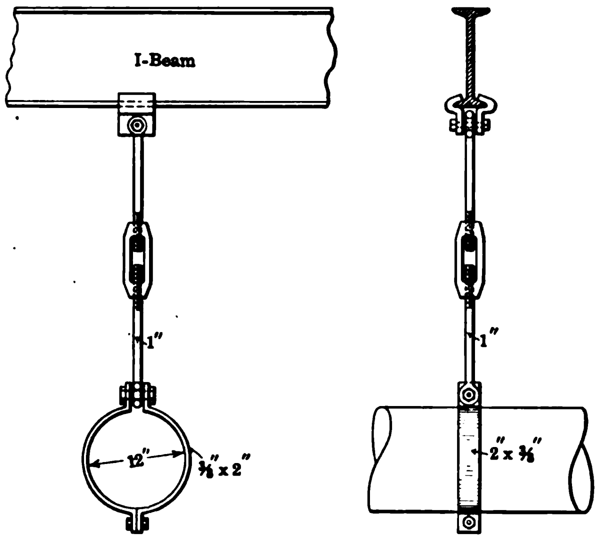


FIG. 82.—Hanger for large pipes.[1]

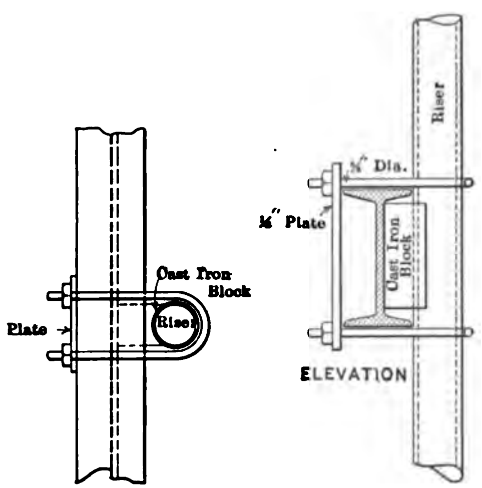


FIG. 83.—Anchor for riser.[1]

[1] From "Pipe-fitting Charts" by W. G. SNOW.

112. Return Piping.—Return pipes of any kind of a steam system should be designed with ample provision for expansion as they may at times be heated to steam temperatures. Dry return mains should be given a pitch of at least 1 inch in 10 feet toward the boiler. Wet return mains should also be pitched toward the boiler so that they may be entirely drained of water when necessary. Return pipes should never be buried in the ground without protection. When it is necessary to conceal them the best plan is to arrange them in trenches with removable cover plates. An alternate scheme is to cover them with cylindrical tile with cemented joints which will keep out the water.

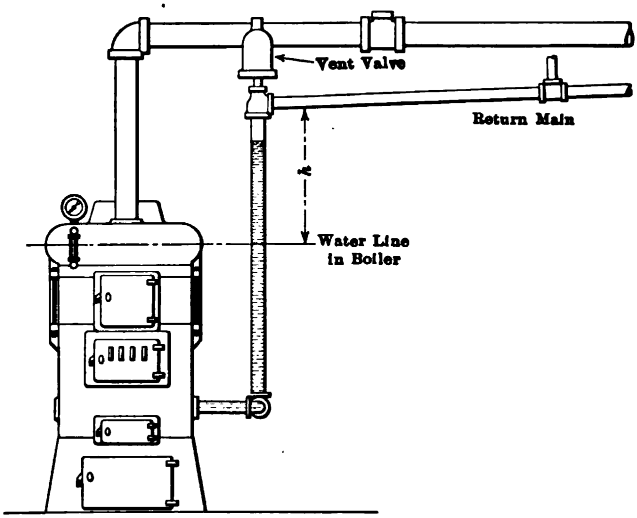


FIG. 84.—Water level in return line of vapor system.

When buried in soil, return pipes corrode and deteriorate very rapidly.

113. Vapor and Vacuum Systems.—In a vapor system, since the return lines are under atmospheric pressure, the water will build up in the return leg (Fig. 84) to a height above that in the boiler equivalent to the pressure in the boiler. In order to prevent the return mains from becoming flooded the distance from the water line of the boiler to the horizontal return main, designated by *h* in Fig. 84, should be as great as possible and should never be less than 2½ feet. In some cases it is necessary to place the boiler in a pit below the basement floor, in order to accomplish

this. The supply main of a vapor system can often be dripped at the end into the return main through a thermostatic trap. This, however, necessitates starting the return main at an elevation below the end of the supply main which, with the necessary pitch toward the boiler, may bring it very close to the water line. A better arrangement is to install a separate drip line from the end of the supply main, which allows the return main

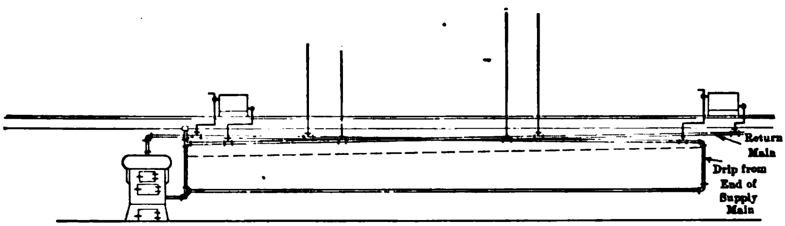


FIG. 85.—Method of dripping supply main when basement is shallow.

to be placed much higher. This arrangement is shown in Fig. 85, the dotted line representing the necessary elevation of the return main if the drip line is omitted.

In an overhead vapor or vacuum system each riser is dripped at the bottom through a thermostatic trap as in Fig. 86. In order to catch the dirt and scale which would clog the trap a dirt pocket should be provided, consisting of a short capped pipe.

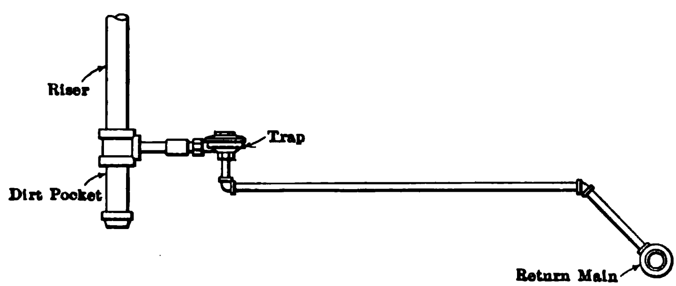


FIG. 86.—Drip connection to riser, vapor or vacuum system.

Steam mains are dripped into the return line in a similar manner. Bypasses are sometimes provided for the most important traps to enable them to be easily cleaned or inspected and dirt strainers are also sometimes used.

114. Valves.—The location of valves in a heating system should be given careful consideration. While valves are desirable

in many locations, there are also some places where they should never be used unless the plant is in the hands of a competent engineer, because of the possibility of accidents resulting from ignorant handling of them.

In a small system as few valves should be installed as possible. Indeed for residence systems it is seldom necessary to install any valves except at the radiators. Valves should never be installed on the steam outlet of the boiler or on the return connection unless the plant is under careful supervision or unless two boilers are used in parallel, in which case valves are necessary in order to enable one boiler to be cut out of service for repairs.

In large buildings a valve should be provided on each riser, if possible, so that the riser can be shut off for repairs, etc., without necessitating the shutting down of the entire system. Valves should also be provided on each branch main and return line in such buildings. Gate or angle valves should be used in preference to globe valves.

115. Radiator Connections.—The connections to a radiator must be sufficiently flexible so that the main or riser is perfectly

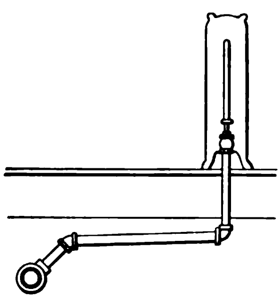


Fig. 87.—Connection to first floor radiator.

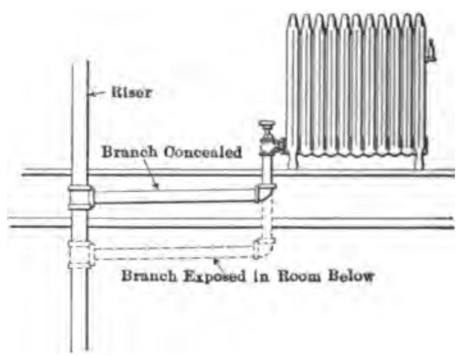


Fig. 88.—Connections from risers where vertical movement is small.

free to expand without straining the fittings. They must also be designed to allow the radiator to drain properly and must be free from water pockets. Figs. 87, 88, and 89 show some proper methods of connecting radiators in a single-pipe system. That shown in Fig. 87 is used for first-floor radiators connected directly to the main. The connection in Fig. 88 is suitable for risers whose vertical movement is small enough to be absorbed by the spring of the horizontal pipe. An objection to this ar-

rangement is the fact that the connection is under the floor and inaccessible unless the horizontal branch is exposed in the room below as shown by the dotted lines. In the connection shown in

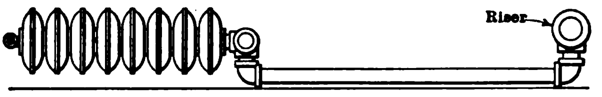


Fig. 89.—Flexible connection, plan view—used when riser has considerable vertical movement.

Fig. 89 a radiator valve of the "corner" pattern is used and the use of the elbows gives a very flexible combination which is well

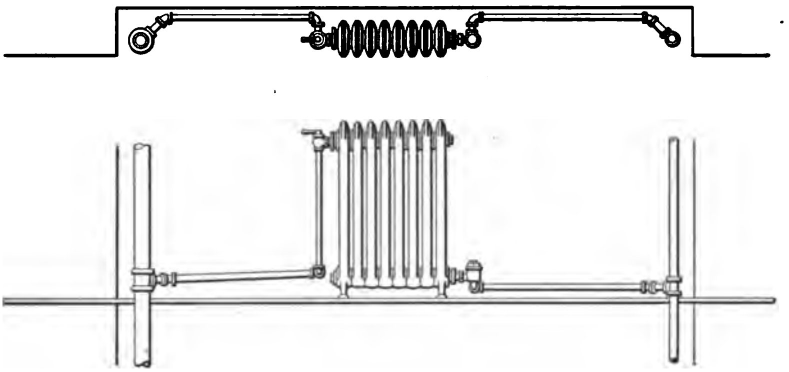


Fig. 90.—Radiator connections—vapor system.

suited for tall buildings where the movement of the risers is considerable.

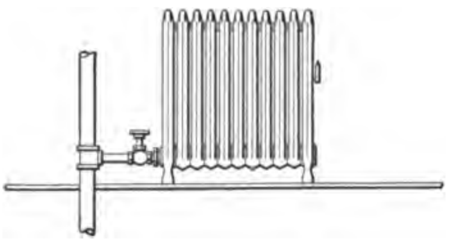


Fig. 91.—Wrong method.

The connections to a radiator of a vapor system are shown in Fig. 90. These connections are also very flexible and the use of 45-degree elbows reduces the frictional resistance.

In no case should a radiator be connected as in Fig. 91. The short, stiff connection allows no free vertical movement of the riser and causes severe strains on the fittings.

116. Pipe Coils.—Pipe coils may be connected in the manner shown in Figs. 92*a* and 92*b*. The arrangement in Fig. 92*a* is used for a two-pipe system and that in Fig. 92*b* for a single-pipe system. A return connection is always used on pipe coils because of the difficulty of draining the large amount of condensation formed in radiation of this type back through the inlet connection. The check valve in Fig. 92*b* prevents steam from entering the coil through the return connection. In order to open the check valve against the pressure of the steam in the riser a water head must be built up above it equivalent to the drop in pressure through the coil, which may be quite appreciable.

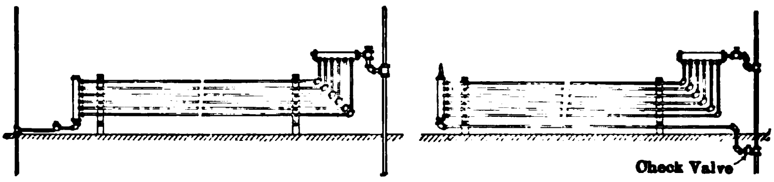


Fig. 92*a*. Fig. 92*b*.
Methods of connecting pipe coils.

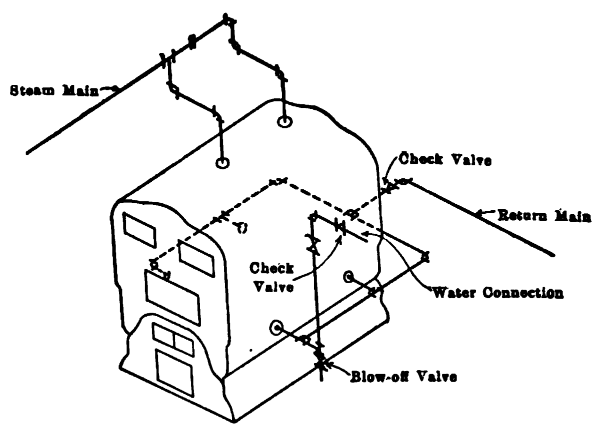


Fig. 93.—Boiler connections.

Therefore, a short length of vertical pipe should be installed above the check valve as shown, to receive the water column which would otherwise occupy the lower part of the pipe coil.

117. Boiler Connections.—The usual method of arranging the connections to a steam boiler is shown in Fig. 93. In

addition to the supply and return connections there is required a blowoff cock and a city water connection with a shutoff valve and a check valve. It is sometimes necessary to connect two boilers in parallel. This must be carefully done so that there will be no chance of either boiler losing water to the other. Connections of ample size between both steam and return connections should be made so that the pressure and water levels in both boilers will be always the same.

118. Flow of Steam in Pipes.—When any fluid flows through a pipe a certain pressure is necessary to move it against the resistance caused by the friction of the fluid against the inner surface of the pipe. The following laws governing the friction of fluids in pipes have been established by experiment:

1. *The total amount of frictional resistance is independent of the absolute pressure of the fluid against the pipe wall.*

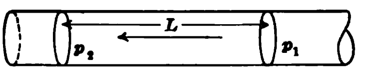


Fig. 94.

2. *The frictional resistance varies nearly as the square of the velocity.*

3. *The frictional resistance varies directly as the area of contact between the fluid and the pipe wall.*

4. *The frictional resistance varies directly as the density of the fluid.*

Consider a condition of steady flow in a pipe and let p_1 (Fig. 94) be the unit static pressure of the fluid, at one point and let p_2 be the pressure at another point at a distance L from the first. The drop in pressure due to the friction of the fluid in passing through the distance L is then

$$P = p_1 - p_2$$

Expressing the laws of friction stated above in algebraic form we have

$$Pa = FSDv^2 \tag{1}$$

in which

P = drop in unit pressure in pounds per square foot.
a = cross-sectional area of the pipe in square feet.
F = a constant depending on the nature of the fluid and the nature of the pipe surface.
S = area of contact between the fluid and the pipe in square feet.
D = density of the fluid in pounds per cubic foot.
v = velocity of the flow in feet per second.

Then $$P = \frac{1}{a} F S D v^2 \qquad (2)$$

Let F be made arbitrarily $= \frac{f}{2g}$

Then equation (2) becomes

$$P = \frac{1}{a} f S D \frac{v^2}{2g} \qquad (3)$$

This is done simply to bring into the expression the term $\frac{v^2}{2g}$ which is the usual form for expressions of this nature.

For round pipes of diameter d and length L, $S = \pi d L$ and $a = \frac{\pi d^2}{4}$.

Then $$P = \frac{4 f L D v^2}{d 2g} \qquad (4)$$

Let w = the weight of steam flowing in pounds per minute.

Then $$w = \frac{\pi d^2}{4} \times v \times D \times 60 = 47.12 d^2 v D$$

and $$v = \frac{w}{47.12 d^2 D} \qquad (5)$$

Let p be the pressure drop in pounds per square inch $= \frac{P}{144}$ and let d_1 be the diameter in inches $= 12d$.

Substituting in (4) these values for v, P and d we have

$$p = 0.04839 \frac{f w^2 L}{D d_1^5} \qquad (6)$$

The coefficient f was found by Unwin to be $= K\left(1 + \frac{3}{10d}\right) = K\left(1 + \frac{3.6}{d_1}\right)$.

The value most commonly used for K for steam is that determined by Babcock which $= 0.0027$.

Substituting in (6) we have

$$p = \frac{0.0001306\, w^2 L \left(1 + \frac{3.6}{d_1}\right)}{D d_1^5} \qquad (7)$$

in which

p = pressure drop in pounds per square inch.
w = weight of steam flowing in pounds per minute.
L = length of pipe in feet.
d_1 = diameter of pipe in inches.
D = average density of steam in pounds per cubic foot.

The value of the coefficient f given above has been found to be correct for small pipes and comparatively low velocities. For large pipes and high velocities the value of f is considerably lower.[1]

119. Factors Affecting the Size of Pipes.—The sizes of pipes to be used in a heating system depend upon several factors. The fundamental requirement as regards the supply pipes is that they must be of sufficient capacity to transmit the required quantities of steam with the pressure differential which is available. The latter depends somewhat upon the source of the steam supply. When exhaust steam from an engine or turbine is used for heating, it is best, from the standpoint of economy, to make possible the carrying of a low back-pressure by designing the heating system to operate with an initial pressure of not over 2 pounds per square inch. The same practice should usually be followed when steam is taken direct from a boiler, as it may be desired at some future time to use exhaust steam. The circulation will also be much better and the system more satisfactory if the pipe sizes are ample. When a vacuum pump is used the greater pressure differential thus set up makes possible the use of smaller pipes but it is well, nevertheless, to design the supply piping to operate as a gravity system with a moderate pressure drop so that the pump can be shut down at times if desired. The return pipes, however, can be made somewhat smaller if a vacuum pump is to be used. Another factor which makes an extreme reduction in the size of the supply pipes undesirable is the noise caused by the resulting high velocity of the steam flowing through them. On the other hand, to make the pipes of excessive size increases unnecessarily the cost of the system. From a consideration of these various factors and of modern practice, a safe standard for the rate of pressure drop in the supply piping may be taken as from 0.03 to 0.10 pounds per 100 feet of pipe.

There are other factors beside that of pressure drop which affect the size of the supply pipes, such as the provision for the carrying of condensation. In general all steam pipes in which the condensation drains in the opposite direction to the flow of steam should be larger than if both flow in the same direction. This

[1] See "The Transmission of Steam in a Central Heating System" by J. H. Walker, *Trans.* A. S. H. & V. E., 1917.

applies particularly to single-pipe radiator connections and branches and to the risers of single-pipe systems.

The proper size of return pipes is based upon experience and good practice as there is no definite law upon which their size can be computed. They must first of all be sufficiently large to carry the condensation. Second, they should be large enough so that they will not become plugged with dirt; and third, they must, in a vapor or vacuum system, be large enough to handle the air from the radiators as well as the condensation, when the radiators are first turned on.

120. Selection of Sizes of Supply Pipes.—In a large or important system it is very desirable to make a detailed calculation of the pressure drop through the system. Besides insuring ample pipe sizes this will enable the pipe sizes to be reduced in some cases below those which would be chosen arbitrarily. In a large building a considerable saving may be effected by judiciously choosing the pipe sizes for the risers and mains. In a vapor system the ideal condition would be to have approximately the same pressure at all radiator valves. To accomplish this fully would be of course an impossibility, but such a condition can be approximated by careful design. In selecting the pipe sizes by the "exact" method, the desired pressure drop through the system is chosen and the approximate average drop per unit length of pipe is found, after which the exact drop can be computed by means of formula (7), Par. 118. In order to facilitate the calculations, the logarithmic chart in Fig. 95 has been prepared, from which the pressure drop per 10 feet of pipe can be read directly. The chart is based on an average density of the steam corresponding to a pressure of 2 pounds gage, which is sufficiently accurate for the range of pressure which occurs in a heating system. In figuring the capacities of the pipes no allowance need be made for condensation in the pipes themselves as this will ordinarily be negligible if the pipes are covered, but if the pipes are to be left bare their radiating surface should be included with that of the radiators. The scales at the bottom of the sheet read directly in square feet of radiation having an assumed heat transmission of 245 B.t.u. per square foot per hour, which is the amount which would be transmitted from 38-inch, two-column radiation with a room temperature of 70° and a steam temperature corresponding to the pressure of 2 pounds. The scales at the top of the sheet read in B.t.u. delivered per hour,

and are convenient for use when the B.t.u. to be delivered by each radiator is known. As an example of the use of the chart, consider a riser 218 feet long supplying 3000 square feet of radiation. If the drop through the riser is to be not more than 0.1 pound, find the proper pipe size. The drop of 0.1 pound in 218 feet is equivalent to a drop of 0.0046 pound in 10 feet. Passing vertically from the 3000-square feet point on the horizon-

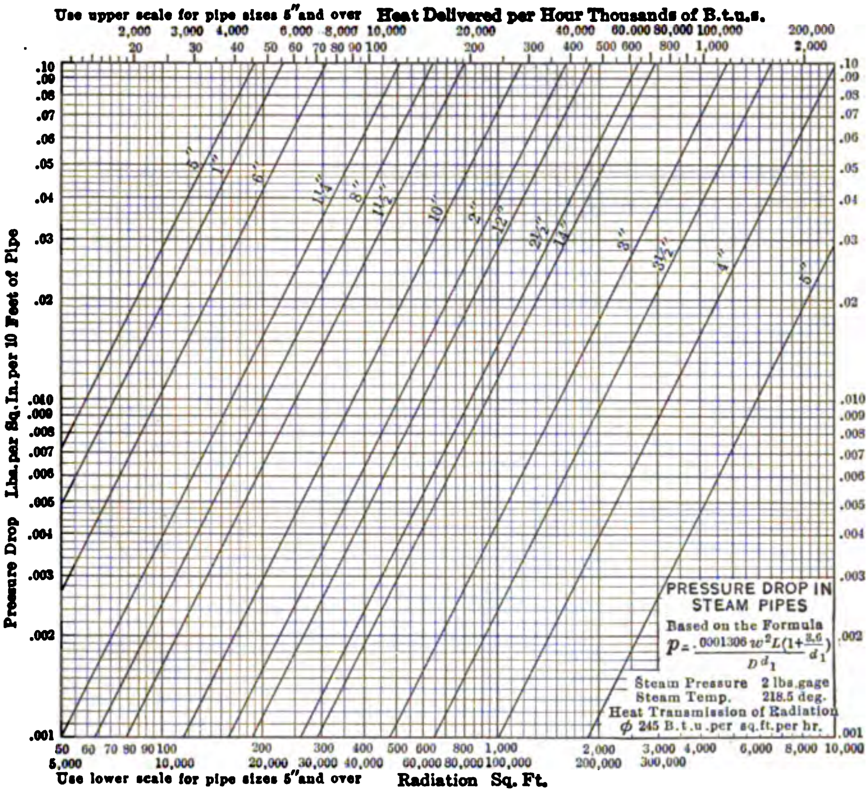


Fig. 95.

tal scale to intersect the diagonal lines for the 4-inch and 5-inch pipes we see that a 5-inch pipe will transmit the steam with a drop of 0.0026 pound in 10 feet and the 4-inch pipe with a drop of 0.0089 pound in 10 feet, which indicates that the 5-inch pipe is the proper size.

The frictional resistance of the fittings must also be considered. It is customary to reduce these resistances to equivalent lengths

of straight pipe, to be added to the actual length, according to the following table.

TABLE XXVII.—EQUIVALENT RESISTANCE OF FITTINGS

Fitting	Equivalent length of straight pipe expressed in no. of pipe diameters
90-degree elbow	40
45-degree elbow	20
Tee	40
Reducing coupling	40
Valve	60

121. Example of Exact Method.—Consider the overhead vapor system shown diagrammatically in Fig. 96, and let it be required to choose the pipe sizes so that the pressure drop through the system will be between 0.3 and 0.5 pound. The equivalent length of each section of pipe should first be computed and set

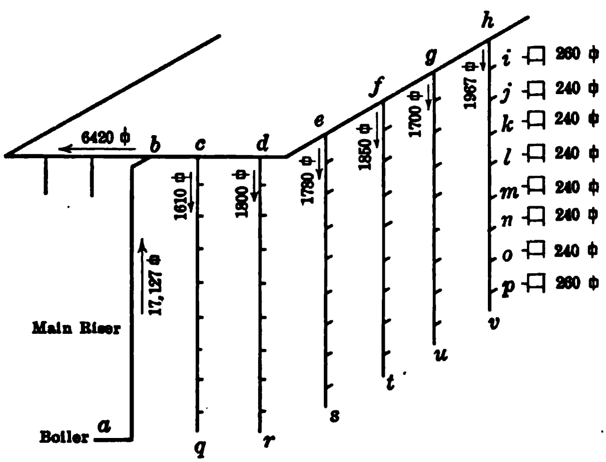


FIG. 96.

down in tabular form. Assuming a pressure of 2 pounds at the boiler, the pressure drop through each section of the main and the riser *h-p*, the longest path of the steam flow, is computed. The total length of the path being 387 feet, the average pressure drop may be taken as $0.4 \div 38.7 = 0.010$ pound per 10 feet of pipe. The pressure drop through each of the successive sections may then be computed from the chart in Fig. 95, using

the pipe sizes which will give as nearly as possible the average pressure drop determined above. The results may be arranged in tabular form as in Table XXVIII.

TABLE XXVIII

Section	Equivalent length, ft.	Rad. supplied	Initial pressure	Pipe size	Pressure drop in section
a–b	130	17,120	2.000	8	0.0075×13.0=0.0974
b–c	20	10,700	1.903	6	0.0125× 2.0=0.0250
c–d	23	9,090	1.878	6	0.0090× 2.3=0.0210
d–e	19	7,290	1.857	5	0.0150× 1.9=0.0290
e–f	27	5,510	1.828	5	0.0090× 2.7=0.0240
f–g	23	3,660	1.804	4	0.0130× 2.3=0.0300
g–h	25	1,960	1.774	3	0.0170× 2.5=0.0420
h–i	15	1,960	1.732	3	0.0170× 1.5=0.0250
i–j	15	1,700	1.707	3	0.0130× 1.5=0.0190
j–k	15	1,460	1.688	3	0.0090× 1.5=0.0140
k–l	15	1,220	1.674	2½	0.0220× 1.5=0.0330
l–m	15	980	1.641	2½	0.0140× 1.5=0.0210
m–n	15	740	1.620	2	0.0220× 1.5=0.0330
n–o	15	500	1.587	2	0.0100× 1.5=0.0150
o–p	15	260	1.572	1½	0.0110× 1.5=0.0160
	387				

Final pressure at p = 1.556 pound.
Total drop = 0.444 pound.

In systems of this kind it is desirable to have about the same pressure at all of the lowest radiators. The other risers, therefore, can be designed for such a pressure drop that the pressure at the bottom of each will be approximately 1.556 pound.

122. Approximate Method.—While the method outlined in the preceding paragraphs should be used for large or important installations, it is quite sufficient for many cases, to choose the pipe sizes simply from the amount of radiation supplied. In Table XXIX are given sizes of mains and return lines for various amounts of radiation for all classes of systems.

123. Radiator Connections.—In order to allow the condensation to drain out against the inflowing steam the connections to radiators of one-pipe systems should be of ample size and the size of the nearly horizontal branches should be still more generously proportioned. In two-pipe systems the radiator supply connections carry little condensation and may therefore be rela-

tively small. The sizes of connections commonly used for radiators of various capacities are given in Table XXX.

TABLE XXIX.—PIPE SIZES FOR SUPPLY AND RETURN LINES

Pipe size	½	¾	1	1¼	1½	2	2½	3	3½
Supply mains—all systems downfeed risers, all systems			50	100	175	350	600	1,000	1,500
Upfeed risers[*]—one-pipe system				50	100	200	300	500	700
Dry return lines—two-pipe and vapor systems		50	150	300	900	2,000	3,800	6,000	10,000
Wet return lines			2,000	3,800	6,000	13,000	23,000	37,000	55,000
Vacuum return lines	100	400	800	1,500	3,000	6,000	10,000	18,000	30,000

Pipe size	4	5	6	8	10	12	14	16
Supply mains—all systems downfeed risers, all systems	2,000	3,800	6,000	13,000	23,000	35,000	55,000	78,000
Upfeed risers[*]—one-pipe system	800	1,300	1,800	3,000				
Dry return lines—two-pipe and vapor systems	13,000	23,000	37,000	78,000				
Wet return lines	78,000							
Vacuum return lines	40,000	65,000						

[*] Which carry condensation from radiators.

TABLE XXX.—SIZE OF RADIATOR CONNECTIONS

One-pipe radiators			Two-pipe radiators		
Size of radiator, square feet	Radiator connection	Horizontal branch	Size of radiator, square feet	Size of supply connection	Size of return connection
20	1	1	48	1	¾
24	1	1¼	96	1¼	1
40	1¼	1¼	over 96	1½	1¼
60	1¼	1½			
80	1½	1½			
100	1½	2			
200	2	2			

Vapor and vacuum systems-supply ¾ inch, return ½ inch.

The size of pipe actually required to convey the necessary amount of steam is usually considerably less than these arbitrary sizes.

124. Erection and Installation of Piping.—It is very necessary that the installation of a heating system be supervised carefully, as an immense amount of trouble can be caused by careless workmanship.

One of the most important points is the proper threading and making up of the pipe joints. Sharp clean threads of the proper length should be the aim, the cutting of which requires that the threading dies be kept in perfect condition. In making up the joints the threads should be wiped perfectly clean and coated with a very small amount of pipe-joint compound. The use of too great a quantity of compound is a frequent and a serious mistake as the substance clogs the traps, valves, and return lines and is a continual source of trouble.

Pipes of the 3-inch size and under are cut with a hand cutter which leaves a burr on the inside of the pipe. In the smaller pipes, especially, a considerable reduction in the internal diameter may thus be produced and the burr should therefore be removed with a reamer.

The piping should be uniformly pitched and all air or water pockets should be avoided. Hangers should be installed in sufficient numbers and in proper locations so that no strains on fittings, valves, or boiler connections will be caused by the weight of the piping.

One common source of trouble especially in new installations is the dirt which gets into the pipng while it is being installed. This dirt, consisting of cement, plaster, chips, etc. from the building operations, and chips produced in threading the pipe, causes a great deal of damage in clogging the pipes, traps, and fittings and in cutting out the valve seats and discs. Most important of all, the open ends of the piping during installation should be kept carefully covered to prevent dirt from entering. Systems having traps on the radiators should always be operated for a week or two without the traps so that most of the dirt will be washed out before the traps are installed.

125. Heating Systems in Connection with Power Plants.—In designing the piping for a heating system to be operated in conjunction with a power plant, provision must be made, first, to use the exhaust steam for heating, with a means for allowing the excess exhaust to escape automatically to atmosphere, and second, to supply live steam to the heating system during the hours when the heating requirements are in excess of the amount of exhaust

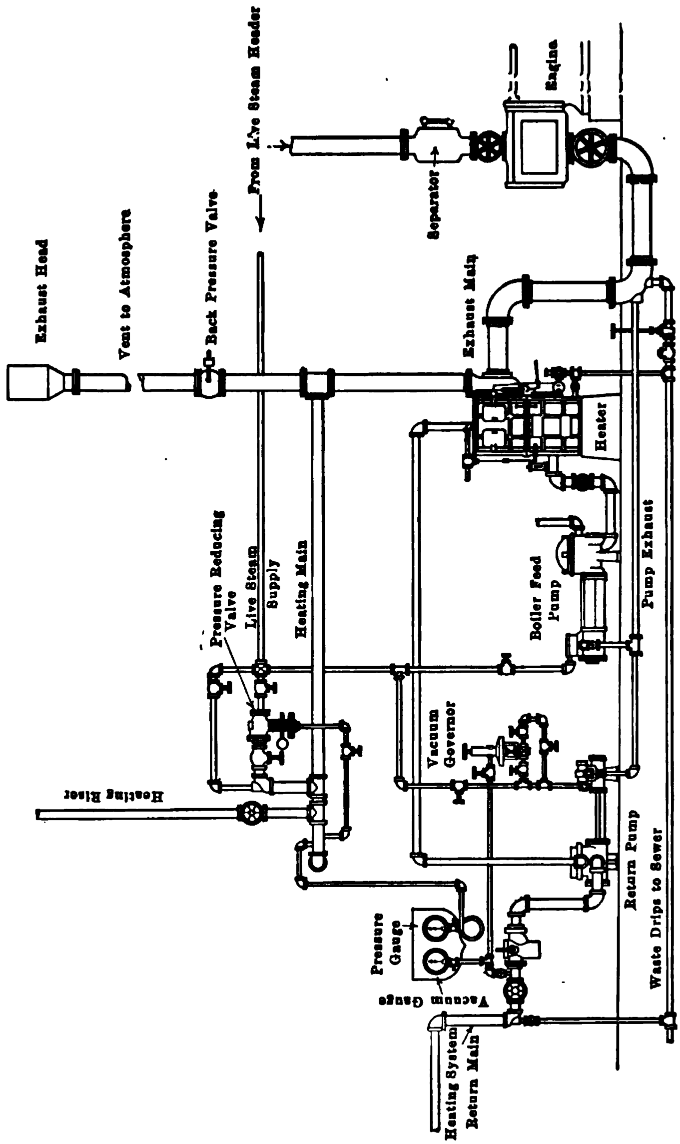


FIG. 97.—Heating system in combination with power plant.

steam available. A common arrangement is that shown in Fig. 97. The back-pressure valve, located on the main exhaust line, is so constructed that an increase of pressure over the amount for which the valve is set causes it to open and discharge steam to the atmosphere. The condensation from the radiators is discharged by the vacuum pump to the open feed-water heater from which it is taken by the boiler feed pump. A pressure-reducing valve with a bypass is used to feed steam direct from the boilers into the heating system when required. The reducing valve may be set to open when the pressure in the heating system, because of an insufficiency of the exhaust steam supply, drops below the required point. The exhaust steam from the pumps is discharged into the main exhaust line, which, it will be noted, has a direct connection to the feed-water heater.

Problems

1. How much steam can be transmitted by a 6-inch pipe 93 feet long with an initial pressure of 5 pounds gage and a final pressure of 4 pounds gage?

2. How much steam can be transmitted by the same pipe as in Prob. 1, with an initial pressure of 105 pounds gage and a final pressure of 104 pounds gage?

3. What will be the drop in pressure if 2000 pounds of steam per hour are passed through a 5-inch pipe, 87 feet long, containing three 90-degree elbows?

4. What initial pressure will be required if 110 pounds of steam per minute flows through a 4-inch pipe 70 feet long, the final pressure being 51 pounds gage? Pipe has two 90 degree elbows.

CHAPTER X

HOT-WATER SYSTEMS

126. Classification of Systems.—In a hot-water heating system the water flows in a closed circuit, absorbing heat while passing through the heater and giving up heat while in the radiators. The force required for moving the water through the circuit may be obtained from either of two sources. In the gravity or "natural" system, the force producing circulation is due to the difference in weight of the hot water in the supply pipes and the cooler water in the return pipes; in the second class of systems the circulation is produced by means of a pump.

Gravity systems are installed in residences and other buildings of moderate size. Since the force producing circulation in a gravity system is small, the velocities are necessarily low and if a large quantity of water must be circulated, it becomes necessary to use very large pipes. Consequently, in large buildings or in groups of buildings where the heating requirements call for a large volume of water, it is best to employ a pump to produce a more rapid circulation, thereby permitting relatively smaller pipes to be used.

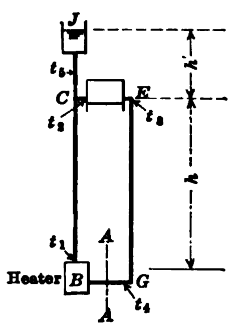


FIG. 98.

127. Gravity System.—Theory of Flow.—As has been previously explained, the force which causes the flow in a gravity or "natural" system is due to the difference in weight of the water in the flow and return pipes. Fig. 98 represents an elementary gravity system, consisting of a boiler and one radiator with an expansion tank.

Consider that the system is in normal operation and that the heat added to the water flowing through the boiler is exactly equal to the heat leaving the water in the radiators and piping. The water leaves the boiler at the temperature t_1 and enters the radiator at the temperature t_2, some heat having been lost during its passage through the pipe BC. In the radiator the water

temperature is reduced to the temperature t_3, and during its passage through the return pipe EG it is further reduced to the temperature t_4, at which temperature it enters the boiler. Let t_5 be the average temperature of the water in the pipe $C-J$ leading to the expansion tank.

Let H be the amount of heat which is delivered per hour by the radiator. Then if Q is the quantity of water flowing in pounds per hour

$$H = Q(t_2 - t_3) \tag{1}$$

The heat lost in the flow piping is

$$H_1 = Q(t_1 - t_2)$$

and in the return piping

$$H_2 = Q(t_3 - t_4)$$

The heat added to the water at the boiler is

$$H' = Q(t_1 - t_4)$$

Then

$$H' = H + H_1 + H_2$$

The density of the water at the various points in the circuit corresponding respectively to temperatures t_1, t_2, t_3, t_4, and t_5 is D_1, D_2, D_3, D_4, and D_5. If the temperature drop is uniform, the average temperature in each section may be taken as the mean of the temperatures at the ends. The average density of the water in BC is then $= \dfrac{D_1 + D_2}{2}$ and in $EG = \dfrac{D_3 + D_4}{2}$.

Now consider the forces acting on each side of the plane $A-A$ passed through the pipe GB. The pressure on the left side is evidently due to the column of water BC of density $\dfrac{D_1 + D_2}{2}$ plus the column CJ of density D_5 which is

$$h\left(\frac{D_1 + D_2}{2}\right) + h'D_5$$

The pressure on the right-hand side is evidently

$$h\left(\frac{D_3 + D_4}{2}\right) + h'D_5$$

Adding these pressures algebraically, we obtain for the resultant pressure tending to move A-A to the left

$$h\left(\frac{D_3 + D_4}{2}\right) - h\left(\frac{D_1 + D_2}{2}\right)$$

Let $$D_F = \frac{D_1 + D_2}{2} \text{ and } D_R = \frac{D_3 + D_4}{2}$$

Then the unit pressure p' available for producing circulation is

$$p' = h(D_R - D_F) \qquad (1)$$

It is evident that this pressure is the same at any point in the circuit *BCEGB*. It is independent of the relative lateral positions of the radiator and the boiler and depends only on the height h and the densities D_R and D_F.

It is customary to express this pressure as a "head," *i.e.*, the height of a column of water of the same density as that in the system which will produce the given pressure at its base. Let D be the average density of the water and h_1 the head equivalent to the unit pressure p'; then $p' = h_1D$ and $h_1 = \frac{p'}{D}$. Substituting in equation (1) we have

$$h_1 = \frac{h(D_R - D_F)}{D}$$

h_1 is then the head available for producing circulation. If D, D_R, and D_F are expressed in pounds per cubic foot and h in feet, then h_1 will be in feet of water column. To express the head in inches, which is a more convenient unit, the right-hand member is multiplied by 12, and

$$h' = \frac{12h(D_R - D_F)}{D} \qquad (2)$$

The density D in equation (2) represents the average density of the water in the system. A close approximation would be to make

$$D = \frac{D_R + D_F}{2}$$

Substituting in (2)

$$h' = 24h\frac{D_R - D_F}{D_R + D_F} \qquad (3)$$

h' is then the available circulating head in inches of water.

128. Friction.—The general expression for the loss of pressure due to friction for fluids in round pipes according to equation (4), page 132, is

$$P = \frac{4fLDv^2}{d2g} \qquad (4)$$

in which

P = loss of pressure due to friction, pounds per square foot.
f = a constant depending on the nature of the fluid and of the pipe wall.
D = average density of the fluid, pounds per cubic foot.
v = velocity, feet per second.
d = pipe diameter, feet.
g = acceleration of gravity = 32.2.
L = length of pipe in feet.

To express the frictional resistance in terms of fluid head, let $P = h''D$ in which P is in pounds per square foot and D in pounds per cubic foot, h'' being the equivalent head in feet of the fluid at density D.

Substituting in (4)

$$h'' = 4f\frac{L}{d}\frac{v^2}{2g} \tag{5}$$

Let $\rho = 4f$, then

$$h'' = \rho\frac{L}{d}\frac{v^2}{2g} \tag{6}$$

Now if v is expressed in inches per second, and d in inches, the head h'' will be expressed in inches of water, without any change in the form of the expression, the inch unit being the more convenient.

Equation (6) gives the frictional resistance to flow through straight lengths of pipe only. The resistance due to elbows and other fittings must also be taken into account. The resistance of such obstructions has been found to be nearly proportional to the square of the velocity of flow, and may therefore be expressed in the form

$$\frac{\alpha v^2}{2g}$$

in which α is a constant to be determined. The summation of all such "single resistances" may then be expressed as

$$\Sigma\alpha\frac{v^2}{2g} \tag{7}$$

and the entire frictional resistance will be

$$h'' = \rho\frac{L}{d}\frac{v^2}{2g} + \Sigma\alpha\frac{v^2}{2g} \tag{8}$$

In order to impart to the mass of water in the system the

velocity v, a certain head must be used up in overcoming this "starting resistance" which is equal to $\frac{v^2}{2g'}$, in which g', the acceleration of gravity, is expressed in inches per second per second so that this last term will be expressed in inches of water head as are the others. The complete expression for the head required to start and to maintain flow may then be written

$$h'' = \rho \frac{L}{d} \frac{v^2}{2g} + \Sigma\alpha \frac{v^2}{2g} + \frac{v^2}{2g'}$$

In which h'' is in inches of water head.

d is in inches.
L is in feet.
v is in inches per second.
g is in feet per second per second.
g' is in inches per second per second.

In considering only the force required to maintain a steady flow, the last term does not enter, however.

129. Condition of Steady Flow.—When the circulation in a heating system has become constant, the head available for producing flow must be exactly equal to the frictional resistance. This condition must invariably be fulfilled. If the available head increases or decreases, the velocity will change also until it assumes such a value that the frictional resistance will equal the available head. The relation[1] may be expressed by equating the right-hand members of equations (3) and (8)

$$24h \frac{D_R - D_F}{D_R + D_F} = \rho \frac{L}{d} \frac{v^2}{2g} + \Sigma\alpha \frac{v^2}{2g} \tag{10}$$

130. Types of Gravity Systems.—Two-pipe Multiple-circuit System.—There are several different ways of arranging the piping in a gravity system. The most common method for installations of moderate size is the two-pipe multiple-circuit system shown in Fig. 99. The water leaves the boiler through the flow main, passes through the radiators and into the return main. A single pair of mains may be installed to circle the basement, but a better method is to install two or more pairs which extend in different directions. In order to insure a

[1] For further discussion see "Heating and Ventilation" by A. H. BARKER, to whom the foregoing analysis is due.

sufficient flow of water to each radiator, it is best to provide separate supply and return risers to each radiator from the mains. Both the supply and return mains are given a pitch toward the boiler of about ½ inch in 10 feet, so that no air will accumulate in the piping and so that the system can be drained at the boiler. Two-pipe systems are often installed with a "reversed" return main, as shown in Fig. 100. The flow in the return main is in the same direction as in the supply main and is so arranged that the length of the circuit through each radiator is the same. This tends to equalize the resistance to flow through all the radiators and the system therefore operates more uniformly throughout.

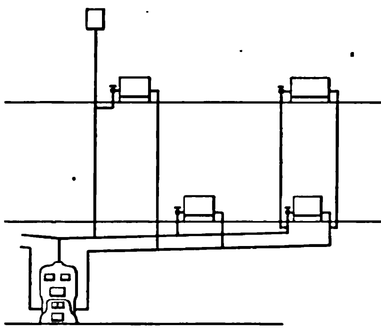


FIG. 99.—Two pipe multiple circuit system.

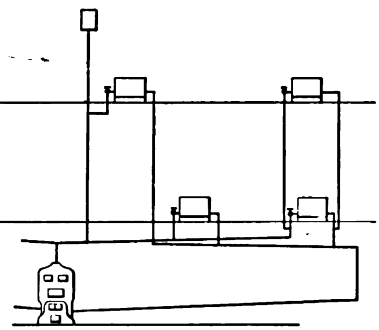


FIG. 100.—Reversed return.

A modification of the two-pipe system was formerly used, in which separate supply and return pipes were provided for each radiator. Although such an arrangement gives good results, the complication and cost of the piping have rendered it practically obsolete.

131. Expansion Tank.—The change of volume of the water in a hot-water system under varying temperatures is quite appreciable and an expansion tank must always be provided.

The tank is located well above the highest radiator in the system and is provided with a vent and an overflow to the sewer, as illustrated in Fig. 101. If located in an unheated room, a connection should be made to it from both supply and return mains to insure sufficient circulation to prevent freezing. If possible, the connection to the tank should be taken from the supply main as near the boiler as possible so that the air which is liberated from any fresh water which is fed to the boiler will rise

to the expansion tank and escape rather than accumulate in the radiators.

The required capacity of the expansion tank is evidently a function of the quantity of water in the system and may be determined by computing the volumetric expansion, for the maximum temperature range, of the estimated quantity of water in the system. A rough rule is to make the capacity of the expansion tank in gallons equal to the radiation in square feet divided by 40.

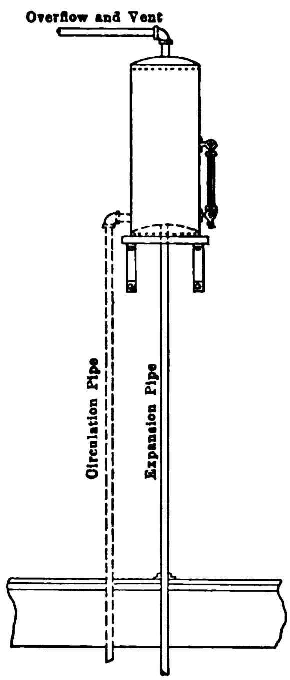


FIG. 101.—Arrangement of expansion tank.[1]

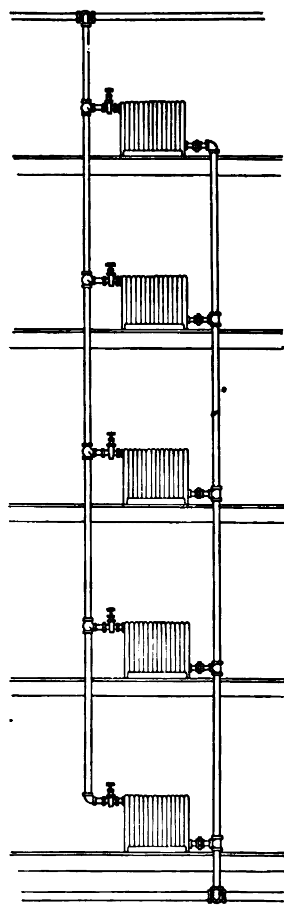


FIG. 102.—Two-pipe overhead system.[1]

132. Two-pipe Overhead System.—In Fig. 102 is shown the two-pipe overhead system. The supply main is located in the attic and parallel supply and return risers drop to the basement as shown. This system is best adapted to rather large buildings.

[1] From "Pipe-fitting Charts" by W. G. SNOW.

133. One-pipe System.—It is perfectly feasible to use a single pipe for both flow and return. A one-pipe overhead system is arranged as shown in Fig. 103. The return line from each radiator is connected to the riser at a point below the supply connection. The circulation through any radiator may be accelerated by lowering the point at which its return connection reënters the riser, as at *B*.

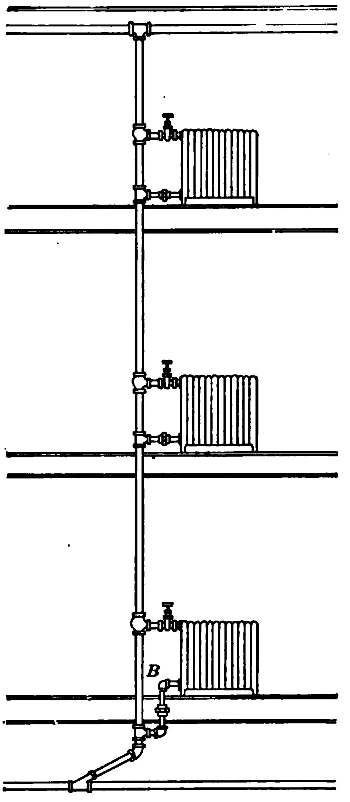


Fig. 103.—One-pipe overhead system.

One disadvantage of this system is the fact that the cool water from the radiators lowers the average temperature of the water in the riser and the radiators on the lower floors are therefore supplied with water at a relatively low temperature, so that they must have a larger surface. The advantages of the one-pipe system are its simplicity and somewhat lower cost.

The one-pipe circuit system is shown in Fig. 104. The main circles the basement and separate connections are usually taken off to each radiator, although a first-floor and a second-floor radiator are sometimes connected to the same risers. The main should be of uniform size throughout its length. In large buildings, a separate main is sometimes installed for each floor. This system has the inherent disadvantage of all one-pipe hot-water systems, that the temperature of

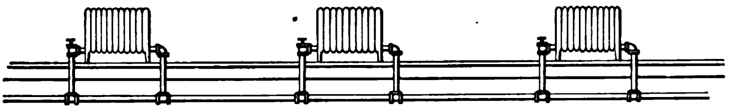


Fig. 104.—One-pipe circuit system.

the water in the main is lowered as that from the radiators is mixed with it and the radiators at the remote end must there-

fore be of larger size. Its chief advantage lies in its simplicity and in the smaller amount of piping required.

134. Water Temperatures.—The water temperatures in a hot-water system will vary according to the heating requirements. Most ordinary gravity systems are designed to operate at a water temperature, leaving the heater, of 180° to 190° and with a drop in temperature through the system of 20° to 30°.

135. Study of Various Types of Systems.—Fig. 105 represents a multiple-circuit system and Fig. 106 an overhead system. The head available for producing circulation through any radiator is proportional to the elevation of the radiator above the boiler, and to the temperature difference between the flow and the return as expressed in formula (3), page 144. In the two types of systems illustrated, the inlet and outlet connections of the radiators are both at the bottom and the effective height should therefore be measured from the radiator connections to the

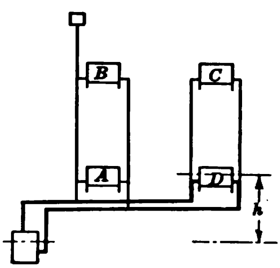


FIG. 105.

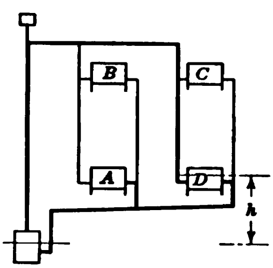


FIG. 106.

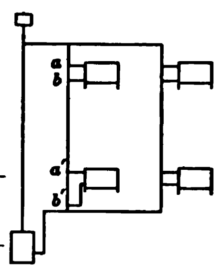


FIG. 107.

center of the boiler. The frictional resistance to flow varies directly as the length l of the circuit from the boiler through the radiator and the circulating head varies directly as the height h of the radiator above the boiler. It is therefore evident that the radiators marked D in both figures are the least favorably situated, since the ratio $\left(\frac{h}{l}\right)$ is the least for these radiators. The size of the pipes in the mains must therefore be based on the circulating head due to these radiators. This can be more clearly comprehended when it is remembered that the source of the circulating force is the radiator itself. Radiators C and D, Fig. 105, may be thought of as centrifugal pumps of different working heads operating in parallel and pumping the water around the circuit. It is evident that in such a case if both

pumps are to deliver water, the force producing circulation could not be greater than that developed by the pump having the smaller head, which corresponds to radiator *D*.

If the pipes are well insulated, the effect of the small amount of heat lost from them will be negligible; if, however, they are left uncovered, the effect on the circulating head will be considerable. In the basement main system, a loss of heat in the flow mains and risers tends to decrease the circulating head, and a loss of heat from the return mains and risers tends to increase it. In the overhead system, a loss of heat from the flow mains and risers as well as from the return piping tends to aid circulation, while a loss from the main riser tends to retard it. This should be evident from a consideration of the direction of flow in these pipes.

136. Single-pipe System.—In the single-pipe system, as illustrated in Fig. 107, the water reaching the inlet connection of a radiator as at *a*, divides, part of the water passing through the radiator and part through the riser from *a* to *b*. The available head for producing flow through the radiator depends upon the distance *a–b* and the difference between the average temperature of the water in the radiator and the water in the pipe *a–b*. A lowering of the point at which the return connection from the radiator enters the riser, as at *b′*, Fig. 107, will tend to cause a greater portion of the water to flow through the radiator.

The circulation through the mains and risers depends upon the lowering of the temperature in the risers themselves. The average temperature in the risers is not necessarily the mean of the temperature at the top and bottom, but depends upon the proportion of the heat removed at the various radiators.

137. Method of Computing Pipe Sizes.—In order to make certain that the system will operate with the same temperature drop and water quantities for which it is designed, it is necessary that the available circulating head be computed from the assumed temperatures and that the pipe sizes be so chosen that the frictional resistance will approximately balance this circulating head. This condition is expressed by equation (10), page 146,

$$24h\frac{D_R - D_F}{D_R + D_F} = \rho\frac{L}{d}\frac{v^2}{2g} + \Sigma\alpha\frac{v^2}{2g}$$

This calculation is, of course, made for the maximum condition. At other times the temperature of the water leaving the boiler,

and consequently the available circulating head, will be less than under maximum conditions.

In Fig. 108 are given the values of the expression $24 \frac{D_R - D_F}{D_R + D_F}$ for various flow and return temperatures. To compute the available circulating head, it is then only necessary to multiply the values obtained from the curves by h, the height of the radiator

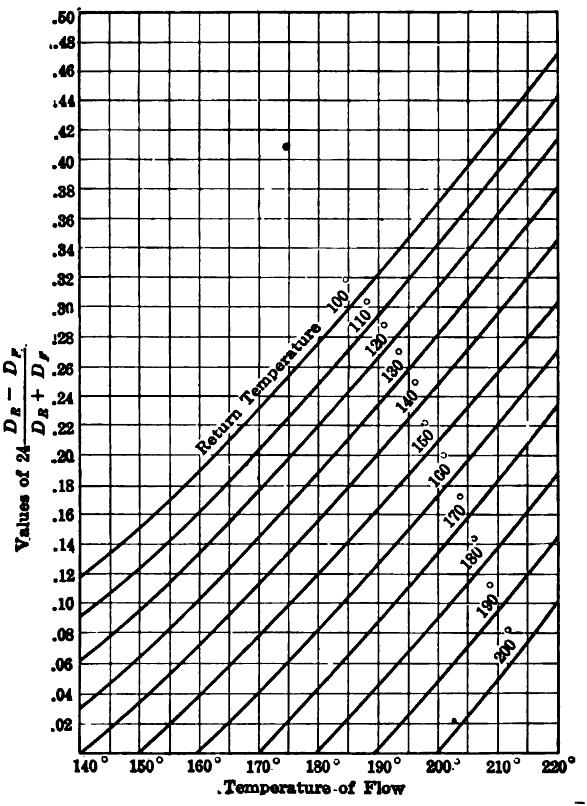


FIG. 108.

above the boiler. The height h should be taken from a point midway between the flow and return connections of the boiler. If both of the radiator connections are at the bottom, the distance h is measured to the connections. If the inlet connection is at the top, the height h is usually measured to a point located at a distance above the bottom connection equal to one-fourth the height of the radiator.

In order to determine the pipe friction, it is necessary to know the value of ρ. This has been determined experimentally by many investigators, but their results differ considerably. According to Weisbach, $\rho = 0.01439 + \frac{0.0595}{\sqrt{v}}$ for water in iron pipes, v being the velocity in inches per second. In order to sim-

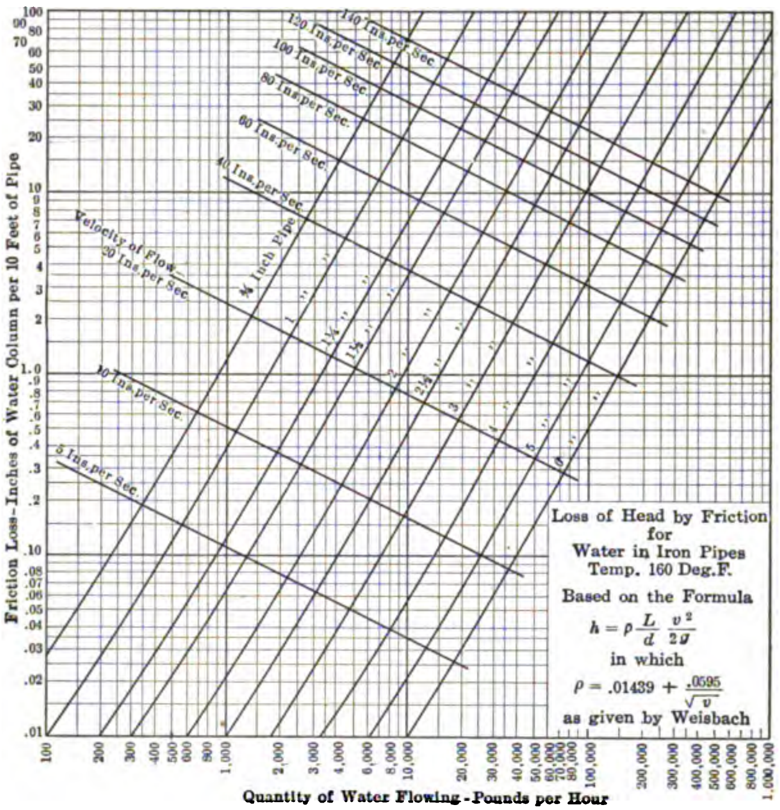


Fig. 109.

plify the determination of frictional resistance under various conditions of flow, the chart in Fig. 109 has been constructed, based on Weisbach's value for ρ.[1] Having given the weight of water

[1] The results of later researches, not fully confirmed, indicate that the Weisbach coefficient is somewhat high and also somewhat in error in that it does not take into account any variation of the friction with the pipe diameter. However, the results obtained from its use are sure to be on the safe

flowing and the pipe size, the resistance in inches of water can readily be taken from the chart.

For the computation of the resistance of the fittings or "single resistances," it is very convenient to consider that the resistance so introduced is equal to that of a certain length of pipe of the same diameter. Approximate determinations of the value of α indicate that at the average velocities occurring in heating work, the length of pipe in feet equivalent to a 90-degree elbow is equal to twice the number of inches diameter of the pipe. For example, a 3-inch elbow is equivalent in resistance to 6 feet of 3-inch pipe. Values for the various single resistance are given in Table XXXI.

TABLE XXXI.—VALUES OF SINGLE RESISTANCES

	Equivalent length in feet equals diameter in inches multiplied by
90-degree elbow	2
90-degree elbow—long sweep	1
45-degree elbow	1
Radiator	4*
Boiler	4*
Valve	1 to 2

* Diameter of pipe connections.

The procedure in calculating the pipe sizes according to the accurate method is then as follows: The piping is completely laid out according to the system chosen, *i.e.*, whether overhead or with basement mains, etc. The circuit supplying the most unfavorably situated radiator is the first to be considered. The pipes in this circuit are assigned tentative sizes and the single resistances noted and the equivalent lengths obtained from Table XXXI. The total equivalent length of each section of the circuit is then computed and the friction drop taken from the curves in Fig. 109. The available circulating head must next be com-

side and it has been used in the design of many successful installations. For further discussion see:

"The Determination of Pipe Sizes for Hot Water Heating Systems," by F. E. GEISECKE, *Trans.* A. S. H. & V. E., 1915.

"The Friction of Water in Iron Pipes and Elbows," by F. E. GEISECKE, *Trans.* A. S. H. & V. E., 1917. "The Mechanics of Heating and Ventilating," by KONRAD MEIER. "Heating and Ventilating" by A. H. BARKER.

puted. From the curves in Fig. 108, the value of $24\frac{D_R - D_F}{D_R + D_F}$ is found for the flow and return temperatures which have been assumed. This value, multiplied by the height in feet of the radiator under consideration, above the boiler, gives the circulating head in inches of water. If the friction head does not agree within about 5 per cent. with the circulating head, as it probably will not in the first calculation, the size of some of the pipes in the circuit must be changed and the total friction drop again computed. By successive refinements the two quantities can be made nearly equal. This circuit having been established, the circuits to the other radiators are worked out in a similar manner, the parts in common with the circuit first computed being left as first set down. In the case of a single-pipe system, the circulation to the most unfavorably situated *riser* is first computed, with the circulating head taken as that due to the riser.

138. Necessity of Accurately Choosing the Pipe Sizes.—Let us examine the effect of an improper selection of pipe sizes. There are three possible ways in which errors can be made.

I. By making all the parts of the system too small but of the proper relative size.

II. By making all of the pipes too large.

III. By making the resistance of some circuits much greater than that in the others.

If the pipe sizes are all too small, the primary effect will be to decrease the quantity of water passed through the entire system in unit time. If the temperature of the water leaving the boiler is kept constant, the effect of the decrease in the quantity will be to increase the temperature drop in the radiators. This will increase the available circulating head which will in turn increase the velocity of flow. Unless the error is extreme, the system will therefore approach the performance set for it.

If the pipes are too large throughout, the primary effect will be to increase the flow of water through the system. This will cause a decrease in the temperature drop through the radiators, a reduction in the circulating head, and a consequent reduction of the flow to some value approaching the proper one. The same action takes place in the case of the individual circuits or radiators. If the pipes are too small, the reduction in flow causes an increase in the temperature drop and the net result is usually but a slight decrease in the heat delivered to the room.

It is thus apparent that gravity hot-water systems are to some extent *self-regulating*. It is due to this property that the ordinary hot-water systems, installed without exact design, operate with satisfaction. Indeed, for the usual small system it is not practicable to make exact calculations of the pipe sizes, experience having evolved "rules of thumb" which give pipe sizes which are on the safe side and produce entirely acceptable results. While the heat delivered to the rooms may vary by several per cent. from the theoretical requirements, the error is well within that due to inaccuracies in computing the heat losses from the room.

In large installations, the exact method has some distinct advantages. The liberality with which the pipe sizes of a small system are selected cannot be practised on a large system without a considerable increase in the cost of the installation, while any pipes which may be chosen too small can be replaced only at great expense. Throttling valves, while they should be placed on the branch circuits as a precaution, are difficult to adjust and are easily tampered with. A calculation of the pipe sizes in the manner outlined is therefore desirable for large or important installations.

139. Approximate Rules for Pipe Sizes.—Table XXXII gives the capacity of mains of various pipe sizes for different kinds of systems.

TABLE XXXII.—SIZE OF MAINS

Assumed Length 100 Feet, Temperature Drop in Radiators 20°

Pipe diam.	Capacity, square feet of direct radiation		
	Two-pipe upfeed	One-pipe upfeed	Overhead
1¼	75	45	130
1½	110	65	190
2	200	121	340
2½	310	190	530
3	540	330	920
3½	780	470	1,330
4	1,100	650	1,800
5	1,900	1,100	3,200
6	3,000	1,800	5,000
7	4,300	2,700	7,200
8	5,900	3,500	9,900

Table XXXIII gives the capacity of risers in square feet of radiation.

TABLE XXXIII.—SIZE OF RISERS

Assumed Temperature Drop in Radiators, 20°

Pipe size	Upfeed				Downfeed risers, not exceeding four floors
	First floor	Second floor	Third floor	Fourth floor	
1	33	46	57	64	48
1¼	71	104	124	142	112
1½	100	140	175	200	160
2	187	262	325	375	300
2½	292	410	492	580	471
3	500	755	875	1,000	810

The following schedule of tappings is used for hot-water radiators:

TABLE XXXIV.—RADIATOR TAPPINGS

Size of radiator	Supply and return tappings
Up to 40 square feet	1 inch
40 to 72 square feet	1¼ inches
Over 72 square feet	1½ inches

140. Piping.—Many of the principles governing the design of steam piping apply to hot-water work. Expansion must be provided for with care, although it is less in amount. Connections and fittings must be installed so as to interpose as little resistance to flow as possible. The venting of the air from the system is important. In addition to a vent at the expansion tank, a small pet-cock should be provided at each radiator and at any other points at which air may accumulate. Mains should be given a pitch of at least ½ inch in 10 feet toward the boiler and provision should be made for draining the water from the entire system as is necessary when the plant is shut down in cold weather.

141. Closed Systems.—In the open-tank systems which have been described, the water temperature is limited to 212° because the pressure at the top of the system is at atmosphere; but if the pressure of the water at the top of the system is raised above atmosphere, its boiling point and consequently the allowable temperature is raised, increasing the heat output of the system For maintaining the increased pressure on the system, some device such as a mercury seal is inserted in the pipe leading to the expansion tank. One form of these so-called "generators" is

shown in Fig. 110. The water from the system, as its temperature rises, exerts an increasing pressure on the surface of the mercury in the chamber *B*, forcing mercury up the tube *A* until it bubbles out of the top of the tube. A pressure equivalent to the height of the mercury column thus formed may be built up at the top of the system and the water may be heated nearly to the corresponding boiling point. As the water in the system cools and decreases in volume, the mercury falls down the tube and more water enters the system from the expansion tank.

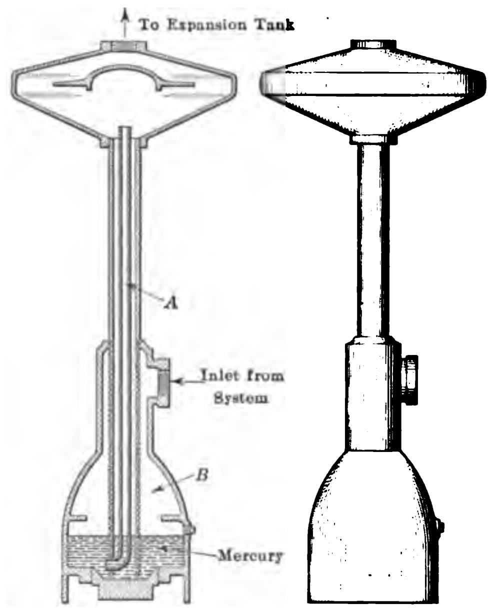


FIG. 110.—Mercury seal "generator."

Generators are especially useful for increasing the output of a heating system which has been inadequately designed or which has become inadequate.

142. Forced Circulation.—When hot-water heating is used in large buildings or groups of buildings, the circulating power is obtained from a pump and smaller pipes are used, the water flowing at much higher velocities than in a gravity system. In systems employing forced circulation, the water usually passes through the pump, then to the heater, and to the radiators. The piping is arranged in the same general manner as in the gravity systems. The action is somewhat different from that in the gravity systems

in that the force producing circulation is from the pump and not from the cooling action of the radiators; for although the temperature difference in the system has some effect, it is so far overbalanced by the force exerted by the pump as to be negligible. The flow through the various parts of the system is therefore governed to a greater extent by the frictional resistance, as the system does not possess the self-regulating qualities of the gravity system.

143. Pumpage, Friction, and Temperature Drop.—The quantity of heat delivered per hour may be expressed by the equation

$$H = Q\ (t_1 - t_2) \tag{1}$$

in which H = quantity of heat delivered per hour.
Q = weight of water pumped per hour.
$t_1 - t_2$ = drop in temperature of water.

It is evident that the quantity of water and the temperature drop may vary, the requirement being that their product remain constant. As the temperature drop is increased, however, the average temperature of the radiators is lowered and somewhat more surface must be installed. It is common practice to allow a temperature drop under maximum conditions of about 20°.

Before a circulating pump can be intelligently selected, it is necessary to choose the differential pressure at which the system is to be operated. If a large pressure drop is allowed, the pipes can be made relatively small, but the power required for pumping the water will be greater. Although it is true that the energy used up in friction is converted into heat and is therefore utilized, the energy thus recovered is only a portion of the energy input to the pumping unit. The cost of the power must therefore be taken into consideration. If the pump is steam-driven and the exhaust used for heating the water, the cost of power will be lower than if current is purchased for a motor-driven pump. In each case a study should be made, balancing the annual investment charges of the piping system against the cost of power to determine the most economical combination. The pressure drop usually allowed is from 10 to 30 pounds. The velocity of flow in the pipes is limited to about 40 inches per second in buildings where the noise produced by a higher velocity would be objectionable. In industrial buildings, no such limit is imposed.

144. Calculation of Pipe Sizes.—The calculation of the pipe sizes in a forced circulation system is much more im-

portant than in a gravity system, because the former does not possess the "self-regulating" property of the gravity system. If any one circuit is unfavorably designed, there will be a tendency for it to be short-circuited. Furthermore, the resistance of the entire system must be made approximately equal to the rated head of the pump. The procedure in designing a forced circulation system is as follows. The heat loss from the building having been computed, the temperature drop in the radiators is chosen and the amount of water to be supplied per hour is com-

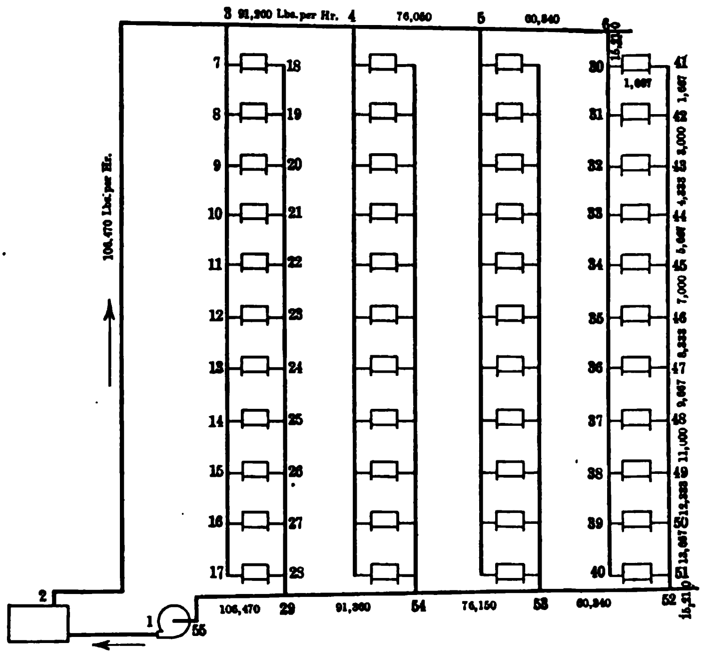


FIG. 111.

puted from formula (1), Par. 143. From a consideration of the various factors mentioned in the preceding paragraph, the differential head is chosen and a pump is selected which will operate most efficiently under the given conditions. The piping must then be designed so that this differential pressure is used up in friction.

The general scheme followed in choosing the pipe sizes is similar to that used for a gravity system, the available circulating head, which in this case is produced by the pump, being balanced by the pipe friction.

The method can best be explained by working out a specific installation. In Fig. 111 is shown diagrammatically one part of an overhead two-pipe system. The weight of water flowing per hour is indicated for the circuit which supplies the radiator marked 30–41, the assumption being made that these water quantities have been computed in the manner previously explained. The circuit through this radiator is the longest and should therefore be computed first and the other parallel circuits designed to give the same resistance. In column 2, Table XXXV, the actual length of each section of the circuit is given. The system will be designed on a basis of a pressure differential of 10 pounds. The length of the circuit is 481 feet. The average

TABLE XXXV.—CALCULATION OF PIPE SIZES—FORCED CIRCULATION SYSTEM

Number of section	Quantity of water flowing, pounds per hour	Proposed diam.	Length of straight pipe	Single resistances	Total equivalent length	Resistance per 10 feet length of pipe	Total resistance	Revised pipe diam.	Single resistances	Total equivalent length	Resistance per 10 feet length of pipe	Total resistance
1	2	3	4	5	6	7	8	9	10	11	12	13
1–2	106,470	4	21	1 × 8	29	4.0	11.6					
2–3	106,470	4	158	3 × 8	182	4.0	72.8					
3–4	91,260	3	22	......	22	9.4	20.7					
4–5	76,050	3	22	......	22	6.8	15.0					
5–6	60,840	3	22	......	22	4.6	10.1	2½	......	22	9.0	19.8
6–30	15,210	2	10	1 × 4	14	2.4	3.4	1½	1 × 3	13	7.5	9.8
30–41	1,667	1	8	2 × 2	12	0.9	1.1					
41–42	1,667	1	12	......	12	0.9	1.1					
42–43	3,000	1	12	......	12	2.8	3.4					
43–44	4,333	1	12	......	12	5.2	6.2					
44–45	5,667	1¼	12	......	12	2.7	3.2					
45–46	7,000	1¼	12	......	12	3.9	4.7					
46–47	8,333	1¼	12	......	12	5.3	6.4					
47–48	9,667	1½	12	......	12	3.3	4.0					
48–49	11,000	1½	12	......	12	4.1	4.9					
49–50	12,333	1½	12	......	12	4.9	5.9					
50–51	13,667	1½	12	......	12	5.9	7.1					
51–52	15,210	2	3	1 × 4	7	2.4	1.7					
52–53	60,840	3	22	......	22	4.6	10.1	2½	......	22	9.0	19.8
53–54	76,150	3	22	......	22	6.8	15.0					
54–29	91,360	3	22	......	22	9.4	20.7					
29–55	106,470	4	29	3 × 8	53	4.0	20.2					
	Total....	.....	.....	....	.....	.....	249.3	.....	......	.....	.....	275.1
	Pounds.	.....	.....	....	.....	.....	8.8	.....	......	.....	.....	9.7

friction loss per 10 feet of pipe in inches of water column at a temperature of 160° will be $\frac{10 \times 1728}{48.1 \times 61.0} = 5.9$ inches of water. With the given quantities of water flowing, and using a friction loss of approximately 5.9 inches per 10 feet, the pipe sizes can be chosen from the chart in Fig. 109, page 153. They are set down in column 3. The length equivalent to the single resistances is computed and the total equivalent lengths set down in column 6. From the friction chart the resistance per 10 feet for each section is found. These are multiplied by the equivalent lengths and the results set down in column 8. The sum of all of them is found to be 249.3 inches of water which is equal to 8.8 pounds as against the 10 pounds originally specified. The sections 5–6, 6–30, and 52–53 may be decreased one pipe size to increase the resistance, as given in columns 9 to 13. The total resistance will then be 275.1 inches or 9.7 pounds which is sufficiently close to the desired resistance. The circuit 2–3–5–53–29–55 and all of the remaining circuits must then be worked out in a similar manner to give an equal resistance, the parts which have already been computed being left as they stand. It is desirable to install a "lock and shield" valve on each riser and at each radiator in order that the distribution can be adjusted after the system is completed.

145. Pumps.—Either the centrifugal or the reciprocating pump may be used to produce the circulation; but the centrifugal type is by far the more suitable. It possesses the advantages of producing a uniform flow of water, does not transmit jars or vibration to the piping, requires little attendance, and is economical in operation. Centrifugal pumps may be driven by either a steam turbine or a motor, the former drive being used when high-pressure steam is available.

CHAPTER XI

AUTOMATIC TEMPERATURE CONTROL

146. Manual Control.—In every heating system the radiators, boiler, and other component parts are selected on the basis of the maximum requirements, *i.e.*, for the lowest outside temperature which is to be expected. Consequently the capacity of the system is much greater than is required in average winter weather. In many localities, for example, where heating plants are designed for a minimum outside temperature of 0°, the average temperature for the heating season is from 35° to 40°. In order to prevent excessive room temperatures the heat output of the system must be regulated, either *manually* or *automatically*, to correspond approximately with the heat losses from the building.

Temperature control is accomplished in different ways according to the kind of heating system and the nature of the building. In many cases manual control of the radiators or of the furnace drafts is all that is necessary; in other cases, automatic temperature control, applied to the individual radiators, is very desirable. In hot-air furnace installations and in small steam and hot-water systems the universal method is to regulate the heat output of the boiler or furnace by adjusting the drafts. When the building is large, however, it is often impossible to regulate accurately the temperature throughout the building by this means and control of the radiators must be resorted to. In vapor systems equipped with graduated inlet valves accurate control is possible if sufficient attention is given by the occupants of the room to the adjustment of the valves. In single-pipe steam systems the supply of steam to each radiator cannot be controlled and it is therefore sometimes desirable to provide at least two radiators in each room so that one or both can be used as required.

In a vacuum steam system the heat output can be varied within certain limits by varying the steam pressure. For example, if the steam pressure could be varied from 10 inches of vacuum to 10

pounds pressure, the temperature of the radiating surfaces would be increased from 193.2° to 240.1°, which, if the room temperature is 70°, would give a range of heat output of about 38 per cent. This is about the maximum range which could be secured by this means.

147. Automatic Control Applied to Boiler or Furnace.—Temperature control by adjusting the drafts of the boiler or furnace can be accomplished automatically by means of any one of several designs of **thermostats.** The simplest of these consists of a bellows containing a volatile liquid which causes an expansion and contraction of the bellows with changes of temperature. The bellows is installed at the point from which the temperature is to be controlled and its movement is transmitted by means of a cable to the dampers on the boiler or furnace in such a way that a lowering of the room temperature causes an increase in the air supply to the fuel bed and a resulting increase in the heat output. This form of thermostat is shown in Fig. 112.

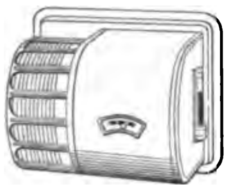


Fig. 112.—Bellows thermostat.

In another form of thermostat the dampers are operated by a motor located in the basement and started electrically from a controller placed in the room above. Fig. 113 illustrates the controller of such a thermostat. The member A consists of two strips of metals, having different coefficients of expansion, brazed together. This member is fixed at point B and the end C is deflected to the right or left by the unequal expansion of the metals with changes of temperature. The controller is connected electrically with the motor in such a way that, as the temperature drops and the strip C makes a contact with D, a current of low voltage is transmitted through the circuit, and, by means of a relay, starts the motor, which opens the drafts on the boiler. Similarly, a slight increase

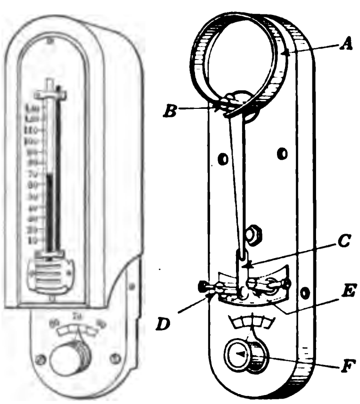


Fig. 113.—Controller for damper thermostat.

of temperature above the established point causes a contact to be made between *C* and *E* and the motor is started, closing the drafts. The temperature for which the controller is set can be changed by moving the knob *F* which shifts the position of *D* and *E*. The controller can be obtained with a clock mechanism which will cause the drafts to close at night and to open in the early morning at some predetermined time.

The motor may be a clock mechanism, in which the energy is obtained from a spring which is wound periodically by hand.

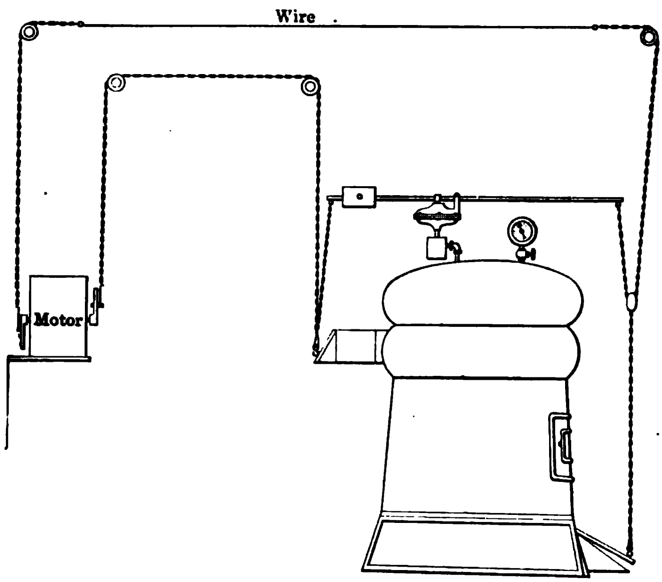


Fig. 114.—Method of connecting thermostat.

The electric motor is more desirable, however, as it requires no winding. The method of connecting the motor to the dampers is shown in Fig. 114.

In installing this form of thermostat the location of the controller is of prime importance. As the heat supply for the entire building is to be controlled from one point, it is essential that the controller be installed in some central location where the temperature is approximately an average of that in the entire building. It is the difficulty of controlling the temperature satisfactorily from a single point that limits the use of such thermostats to residences and small buildings.

148. Automatic Control Applied to Individual Radiators.—In large buildings, in order to regulate the temperature automatically, the radiators in the various rooms must be operated as separate units, by means of a controller located in each room. The power for operating the radiator valves is obtained from compressed air, supplied from a central source, and the air supply to the individual radiator valves is regulated by a small valve operated by the expansion element in the controller. The system may be designed so that the radiator valves are either fully open or fully closed, or the amount of opening may be graduated according to the room temperature. The former arrangement is necessary on single-pipe radiators and is known as the "positive" type, while the latter or "graduated" type is applicable to steam radiators having a separate return connection and to hot-water radiators.

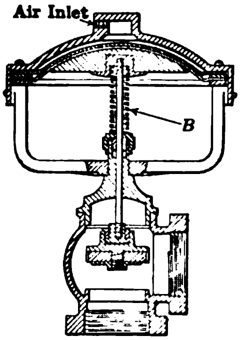


FIG. 115.—Radiator valve for compressed air system of temperature regulation.

The type of radiator valve used is shown in Fig. 115. The valve is closed when air under sufficient pressure is admitted above the diaphragm *A*. When the air pressure is released the springs *BB* force the valve open. If a pressure less than that required to close the valve exists above the diaphragm the valve will take an intermediate position depending on the amount of that pressure. In the graduated system the intermediate positions of the radiator valve are obtained by creating this partial pressure.

A common design of compressed-air thermostat of the graduated type is shown in Fig. 116. The thermostatic element is the hard-rubber cylinder *A*. The valve *G* is closed while the room temperature is up to normal and the full air pressure is transmitted through the inlet *C*, the restricting valve *S*, and the outlet *D* to the diaphragm chamber in the radiator valve, keeping the valve closed. When the tube *A* contracts, due to a lowering of the room temperature, a downward force is exerted on the rod *K* and the block *L*, moving the valve lever *O* to the right against the pressure of the spring *N*, and opening the valve *G* slightly. Because of the restricted passage at *S* the air pressure in the passage *Y* and in the diaphragm chamber of the radiator valve, is lowered, allowing the latter to open and to admit some

steam to the radiator. A further contraction of the tube A causes a further lowering of the air pressure at Y and an increase in the opening of the radiator valve. A thermostat of the positive type is so constructed that an opening of the valve corresponding to G causes a complete reduction of the pressure at Y, allowing the radiator valve to open wide.

149. Compressors.—The air supply is obtained from a small compressor, usually motor-driven, located in the basement. A storage tank is required and a constant pressure is maintained in the tank by means of a governor which automatically starts

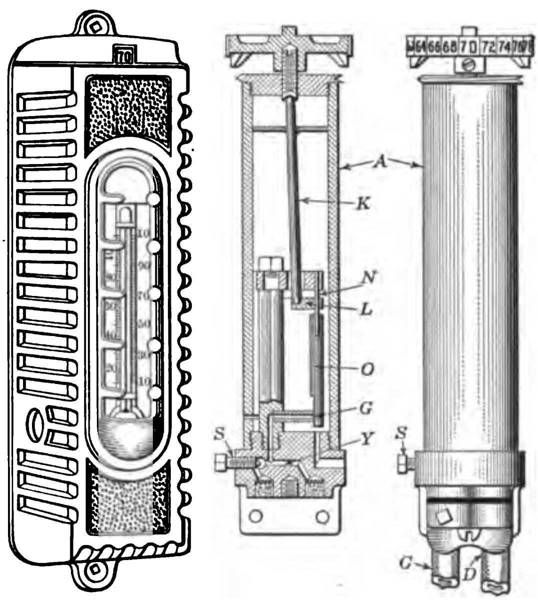


Fig. 116.—Compressed air thermostat-graduated type.

and stops the compressor, as required. The pressure carried on the tank is usually about 25 pounds per square inch.

The mixing dampers and the heating coils of a fan system can be readily controlled by thermostats, through the use of a diaphragm motor as shown in Fig. 117. The control of humidity is also possible by the use of similar devices. These applications will be considered more fully under "Fan Systems."

150. Advantages of Automatic Control.—The advisability of installing a system of thermostatic control depends largely upon the type of building under consideration. The principal advan-

tages of thermostatic control are the convenience and the increased comfort which it affords the occupants. Without any manipulation of the radiator valves, the temperature of the rooms is maintained at the most comfortable point, regardless of the outside temperature. In many cases a considerable saving in fuel can be effected by the use of automatic control, due to the fact that with manual control there is always a tendency for the rooms to become overheated through lack of attention to the radiator valves. This may be true even when graduated valves or other means of facilitating hand control are provided. The

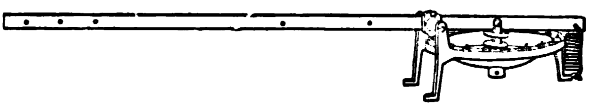


FIG. 117.—Diaphragm motor.

actual amount of the saving in fuel is problematical, being given by many as from 10 to 30 per cent. In the average case it is probable that the lower figure is the more nearly correct.

The objections to the compressed-air systems of thermostatic control are the rather high initial cost of the apparatus and the cost of maintaining and of keeping in adjustment the various parts of the system. Thermostatic control is especially desirable for hotels, schools, office buildings, and other buildings of a public character. For fan systems, automatic control of the dampers and coils is very much to be desired, and in most cases is absolutely necessary if satisfactory results are to be obtained.

CHAPTER XII

AIR AND ITS PROPERTIES

151. Composition of Air.—The atmosphere of the earth is a mixture of several gases and vapors, the proportions of which vary somewhat in different localities and under different weather conditions. In general the proportions of nitrogen and oxygen, the two most important constituents of dry air, are approximately as follows:

	By weight	By volume
Nitrogen......................	76.9	79.1
Oxygen.......................	23.1	20.9

Carbon dioxide and water vapor are also contained in air in varying amounts and there are in addition small quantities of other gases, such as argon, ozone, and neon, which are of less importance. Air is not a chemical combination but is a *mechanical mixture* of these gases.

152. Oxygen.—Oxygen, (O), which constitutes about one-fifth of the air by volume, is the element upon which animal life is dependent for its existence. In the process of respiration the lungs draw in and expel periodically a small quantity of air and a portion of the oxygen unites chemically, while in the lungs, with impurities of the blood, and thereby cleanses it. Some of the resulting products of this chemical reaction are exhaled in the form of gases and vapors. Our health and bodily comfort are dependent upon the proper performance of this process.

153. Nitrogen.—Nitrogen, (N), which constitutes nearly all of the remaining four-fifths of the air by volume, is a relatively inert gas. It performs the important function of diluting the oxygen. As the human body is organized this dilution is essential; an atmosphere of pure oxygen would soon burn up and destroy the body tissues.

154. Carbon Dioxide.—Carbon dioxide, (CO_2), exists in small amounts in the open air, the purest air containing from 3 to 4 parts of CO_2 by volume in 10,000. Carbon dioxide is also known as carbonic acid gas, as it forms a weak acid when dissolved in water.

Being one of the products of respiration it is found in larger quantities in the air of occupied rooms. Carbon dioxide was for a long time believed to have a poisonous effect when taken into the lungs, but is now known to be quite harmless, of itself, even in appreciable amounts. It has the effect, however, of diluting the oxygen content of the air. This necessitates an increase in the rate of breathing and under extreme conditions causes great discomfort. Haldane and Priestly found that with 2 per cent. of CO_2 the lung action was increased 50 per cent.; with 3 per cent. of CO_2 about 100 per cent.; with 4 per cent. of CO_2 about 200 per cent.; and with 6 per cent. of CO_2 about 500 per cent. With 6 per cent. breathing becomes very difficult, while with more than 10 per cent. there occurs a loss of consciousness, but no immediate danger to life. Exposure to an atmosphere containing even 25 per cent. of CO_2 does not result in immediate death.

Being a product of respiration the amount of CO_2 present in the atmosphere of a room is an indication of the amount of air being supplied to the room. The measurement of the CO_2 content of air is therefore of importance in ventilating work. There are several methods of measurement in use, the most accurate of which is that devised by Petterson and Palmquist. The apparatus is provided with a graduated chamber into which a sample of air is drawn and measured. It is then made to flow into a burette containing a saturated solution of caustic potash which absorbs the CO_2. The air is then forced back to the measuring chamber and the decrease in volume noted. The apparatus is calibrated to read directly in parts per 10,000.

Another method sometimes used is that of Wolpert. A solution of sodium carbonate of known concentration is made up and a small quantity of phenolphthalein indicator is mixed with it. A suitable piston arrangement is used to force a known volume of the air to be analyzed into contact with the solution and the apparatus is shaken to promote the reaction between the acid CO_2 and the alkaline solution. The process is repeated several times until the original pink color of the solution disappears. The number of charges of air necessary to cause the color change gives an indication of its CO_2 content.

155. Water Vapor.—Water vapor is an important constituent of the atmosphere. It is the most variable in quantity of any of the atmospheric elements, its amount depending largely

on the weather conditions. In the northern part of the United States the range of the moisture content of the atmosphere is very great. In New York, for example, it varies at different times from 0.5 grain to 7 grains per cubic foot. Water vapor, strictly speaking, is nothing other than *steam at very low pressures*, and its properties are similar to those of steam. This fact should always be borne in mind when dealing with the subject of atmospheric moisture. Another conception that should be thoroughly understood is that of Dalton's law of partial pressures. According to this law, *in any mechanical mixture of gases, each gas has a partial pressure of its own which is entirely independent of the partial pressures of the other gases.* For example, consider a cubic foot of hydrogen gas at an absolute pressure of 5 pounds per square inch. If a cubic foot of nitrogen at an initial pressure of 10 pounds per square inch be injected into the same space, the resulting total pressure will be 15 pounds per square inch and the volume 1 cubic foot. In air, therefore, the oxygen, nitrogen, water vapor, and other gases each have their own partial pressure, the sum of all of them being equal to the total or barometric pressure.

For every *temperature* there is a corresponding *partial pressure* of water vapor at which the vapor is in a **saturated** state, its condition then being exactly similar to that of saturated steam, *i.e.*, with the maximum number of molecules occupying a unit space. When the water vapor is in a saturated condition the air is also spoken of as being saturated since it then contains the maximum weight of vapor which it can hold at that temperature. If the temperature of the air is higher than that corresponding to the partial pressure of the water vapor, the vapor is **superheated**; if the temperature drops below the saturation point some of the vapor is condensed and the vapor pressure is lowered to that corresponding to the new temperature. The saturation temperature is termed the **dew point**. The partial pressure of saturated vapor increases as the temperature increases. Consequently air at higher temperatures is capable of holding a greater weight of water per cubic foot. It should be remembered that *the water vapor exists independently of the air except for the temperature effect of the latter;* and the vapor may be thought of as occupying the given volume at its own partial pressure. The state of intimate mixture of the air and vapor causes their temperatures to be always the same.

156. Relative and Absolute Humidity.—Atmospheric moisture is termed **humidity. Absolute humidity** is the actual vapor content expressed in grains per cubic foot or per pound of air. The ratio of the vapor content to the vapor content of saturated air at the same temperature, expressed in per cent., is called the **relative humidity.** For example, given a sample of air at 70° having an absolute humidity of 4 grains per cubic foot. Since saturated air at 70° contains 8 grains per cubic foot, the relative humidity is 50 per cent.

157. Total Heat of Air.—The **total heat** above 0° of air containing aqueous vapor is the sum of the heat of the air and the heat of the vapor. The latter has three components: the **heat of the liquid,** the **heat of vaporization,** and the **superheat.** In dealing with air containing vapor it is often convenient to use the units of weight instead of volume as a basis for calculations. The total heat above 0° in 1 pound of dry air at temperature t_a is equal to

$$H = C_{pa}(t_a - 0)$$

in which t_a is the air temperature and $C_{pa} = 0.2415$, the specific heat of air at constant pressure.

Let W_w = the weight of water vapor contained in 1 pound of a mixture of air and water vapor. Then for saturated atmosphere

$$H = (1 - W_w) \times C_{pa}(t_a - 0) + W_w(h' + r)$$

in which h' = heat of the liquid above 0° for the water vapor.

r = latent heat of the water vapor.

For atmosphere below saturation at temperature t_a

$$H = (1 - W_w) \times C_{pa}(t_a - 0) + W_w(h' + r + C'_{ps}(t_a - t_d))$$

in which t_d is the temperature at the dew point and C'_{ps} is the specific heat of water vapor at constant pressure.

158. Adiabatic Saturation.—When air below saturation is brought into intimate contact with water there is always a tendency for some of the water to vaporize, adding to the moisture content of the air. If no heat is added from an outside source and none removed, the heat of vaporization for the moisture which is added will be supplied entirely at the expense of the heat of the air and of the superheat of the original quantity of water vapor. The process will continue until the saturation point is reached. A process of this nature taking place without

a transfer of heat to or from an outside source is called **adiabatic** and the final temperature which is reached is therefore termed the temperature of **adiabatic saturation.** Its *depression* below the original temperature of the air *will depend upon the amount of moisture which was added* to bring the air to saturation.

The heat used in the vaporization of the moisture which was added is exactly equal to the heat given up by the air and by the water vapor which it contained originally, assuming that the water which was added was at the temperature of adiabatic saturation. The action may be expressed algebraically as follows:[1]

Let t = temperature of the air.

t' = temperature of adiabatic saturation.

W' = weight of water vapor mixed with 1 pound of dry air at saturation at temperature t'.

W = weight of water vapor mixed with 1 pound dry air at temperature t.

$W' - W$ = weight of water added per pound of dry air.

r = latent heat of vaporization at temperature t.

C_{ps} = specific heat of water vapor at constant pressure.

C_{pa} = specific heat of dry air at constant pressure.

$$(W' - W)r = C_{ps}W(t - t') + C_{pa}(t - t') \qquad (1)$$

or

$$W = \frac{rW' - C_{pa}(t - t')}{r + C_{ps}(t - t')} \qquad (2)$$

159. Measurement of Humidity.—The principle stated in the preceding paragraph affords a convenient means for measuring humidity, through the use of the **wet- and dry-bulb thermometer.** The instrument consists of two mercury thermometers, the bulb of one of which is covered with cotton wicking. The end of the wicking extends into a bottle of water and the entire length is kept wet by absorption. As the water is evaporated from the wicking its temperature is lowered to the temperature of adiabatic saturation or "wet-bulb" temperature. By reading both thermometers when they have reached a constant point the wet-bulb depression is obtained and the moisture content of the air (W) can be found from equation (2), Par. 158.

Distinction should be drawn between the *wet-bulb temperature* and the *dew point,* which was defined in Par. 155. The former

[1] From "Rational Psychrometric Formulæ" W. .H. CARRIER, *Trans.* A. S. M. E., 1911.

temperature is produced by ***adding moisture*** to the air and causing its temperature to drop by reason of the giving up of heat to vaporize the water. The dew point, on the other hand, is reached by ***removing heat*** from the air without changing its moisture content. In order to obtain accurate results it is necessary that the air surrounding the wet-bulb thermometer be in motion so that the maximum evaporation may be secured. For this reason the best form of wet- and dry-bulb thermometer is the "sling psychrometer" illustrated in Fig. 118. In this instrument the wet- and dry-bulb thermometers are mounted on a metal strip pivotted to a handle. In using the instrument the wick surrounding the wet bulb is moistened and the instrument is whirled rapidly and read at intervals until there is no further drop in the wet-bulb temperature. Somewhat more accurate results are obtained with the "aspiration" psychrometer in which a continuous current of air is drawn over the wet-bulb thermometer by means of a small fan driven by clockwork.

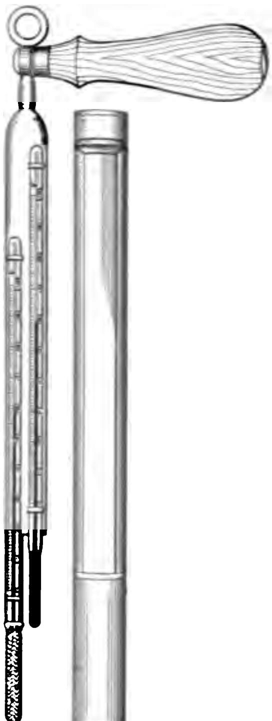


FIG. 118.—Sling psychrometer.

It is necessary that the water used to moisten the wet bulb of the sling psychrometer be at approximately the wet-bulb temperature; otherwise the time required to bring the water to the wet-bulb temperature might be so great that parts of the wicking would become dry. The ideal psychrometric chart in Fig. 119 is constructed for use with the sling psychrometer.[1] This chart gives the moisture content of air in grains per cubic foot, the volume basis being the more convenient for ordinary ventilating work. In Figs. I and II, in the Appendix, are given the psychrometric charts which give the properties of air on the basis of 1 pound of air.

160. Example of Use of Psychrometric Chart.—Given a dry-bulb temperature of 80° and a wet-bulb temperature of 70°,

[1] From "Fan Engineering," Buffalo Forge Company.

find the relative and absolute humidity and the dew point. From the 80° point on the horizontal scale follow the vertical line to its intersection with the diagonal line representing the wet-bulb temperature of 70°. Passing horizontally to the left from this point to the left-hand scale we find that the absolute humidity is 6.65 grains per cubic foot. To find the relative humidity we note that this same point lies between the 60 and 70 per cent. relative humidity lines (the curved lines extending

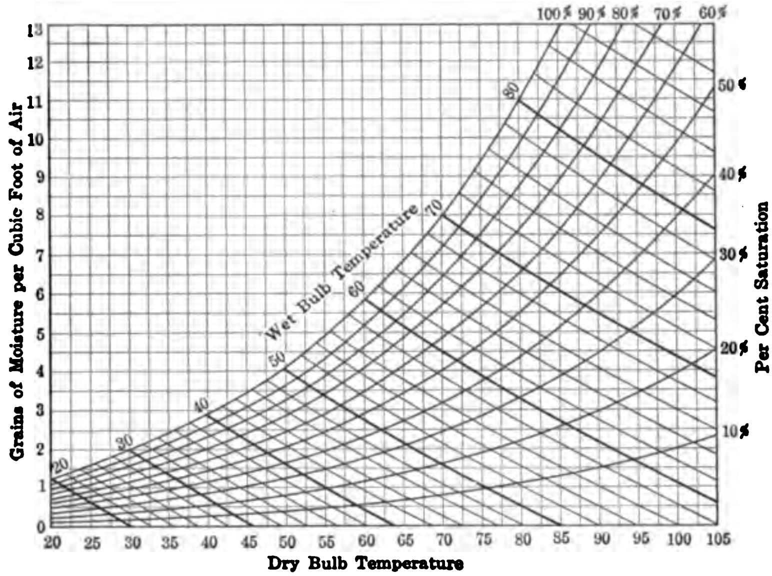


FIG. 119.—Psychrometric chart.

upward to the right) and that the relative humidity is 62 per cent. To find the dew point, follow left horizontally from this same point to the curved line of wet-bulb temperatures, called the saturation line. The dew point is 64.5°.

The relation between the wet- and dry-bulb temperatures and the dew point should be thoroughly understood.

161. Application to Air Conditioning.—If water is sprayed continuously into the path of a current of air and the same water is recirculated repeatedly the temperature of the water will approach the wet-bulb temperature of the air. The latter will not change but the dry-bulb temperature of the air will be lowered until it approaches the wet-bulb temperature, and at saturation the two will coincide. *The wet-bulb temperature depends upon*

the total heat of the air and vapor and will be constant so long as the total heat of the mixture of air and vapor is constant. In the process mentioned the heat of the air above the wet-bulb temperature and the superheat of its original water vapor content go to supply the heat of vaporization for the added moisture, as expressed by equation (1), Par. 158. This means is often employed to cool the air for ventilation.

If a spray of artificially cooled water be used the air can be cooled to within a few degrees of the water temperature. If this temperature is below the dew point of the air some of the moisture content will be condensed and the resulting condition will be one

TABLE XXXVI.—PROPERTIES OF DRY AIR[1]

Barometric Pressure 29.921 Inches

Temperature, deg. F.	Weight per cu. ft., pounds	Ratio to volume at 70° F.	B.t.u. absorbed by 1 cu. ft. dry air per deg. F.	Cu. ft. dry air warmed 1° per B.t.u.	Temperature, deg. F.	Weight per cu. ft., pounds	Ratio to volume at 70° F.	B.t.u. absorbed by 1 cu. ft. dry air per deg. F.	Cu. ft. dry air warmed 1° per B.t.u.
0	0.08636	0.8680	0.02080	48.08	130	0.06732	1.1133	0.01631	61.32
5	0.08544	0.8772	0.02060	48.55	135	0.06675	1.1230	0.01618	61.81
10	0.08453	0.8867	0.02039	49.05	140	0.06620	1.1320	0.01605	62.31
15	0.08363	0.8962	0.02018	49.56	145	0.06565	1.1417	0.01592	62.82
20	0.08276	0.9057	0.01998	50.05	150	0.06510	1.1512	0.01578	63.37
25	0.08190	0.9152	0.01977	50.58	160	0.06406	1.1700	0.01554	64.35
30	0.08107	0.9246	0.01957	51.10	170	0.06304	1.1890	0.01530	65.36
35	0.08025	0.9340	0.01938	51.60	180	0.06205	1.2080	0.01506	66.40
40	0.07945	0.9434	0.01919	52.11	190	0.06110	1.2270	0.01484	67.40
45	0.07866	0.9530	0.01900	52.64	200	0.06018	1.2455	0.01462	68.41
50	0.07788	0.9624	0.01881	53.17	220	0.05840	1.2833	0.01419	70.48
55	0.07713	0.9718	0.01863	53.68	240	0.05673	1.3212	0.01380	72.46
60	0.07640	0.9811	0.01846	54.18	260	0.05516	1.3590	0.01343	74.46
65	0.07567	0.9905	0.01829	54.68	280	0.05367	1.3967	0.01308	76.46
70	0.07495	1.0000	0.01812	55.19	300	0.05225	1.4345	0.01274	78.50
75	0.07424	1.0095	0.01795	55.72	350	0.04903	1.5288	0.01197	83.55
80	0.07356	1.0190	0.01779	56.21	400	0.04618	1.6230	0.01130	88.50
85	0.07289	1.0283	0.01763	56.72	450	0.04364	1.7177	0.01070	93.46
90	0.07222	1.0380	0.01747	57.25	500	0.04138	1.8113	0.01018	98.24
95	0.07157	1.0472	0.01732	57.74	550	0.03932	1.9060	0.00967	103.42
100	0.07093	1.0570	0.01716	58.28	600	0.03746	2.0010	0.00923	108.35
105	0.07030	1.0660	0.01702	58.76	700	0.03423	2.1900	0.00847	118.07
110	0.06968	1.0756	0.01687	59.28	800	0.03151	2.3785	0.00782	127.88
115	0.06908	1.0850	0.01673	59.78	900	0.02920	2.5670	0.00728	137.37
120	0.06848	1.0945	0.01659	60.28	1000	0.02720	2.7560	0.00680	147.07
125	0.06790	1.1040	0.01645	60.79	1200	0.02392	3.1335	0.00603	165.83

[1] From "Fan Engineering," Buffalo Forge Company.

of saturation at the final temperature. These principles are applied practically in the cooling and dehumidifying of air which will be discussed in Chapter XVI.

162. Properties of Dry and Saturated Air.—The properties of dry air are given in Table XXXVI and the properties of saturated air in Table XXXVII, at the standard barometric pressure of 29.92 inches of mercury.

TABLE XXXVII.—PROPERTIES OF SATURATED AIR[1]

Weights of Air, Vapor of Water, and Saturated Mixture of Air and Vapor at Different Temperatures, Under Standard Atmospheric Pressure of 29.921 Inches of Mercury

Temperature, deg. F.	Vapor pressure, inches of mercury	Weight in a cu. ft. of mixture			B.t.u. absorbed by 1 cu. ft. sat. air per deg. F.	Cubic feet sat. air warmed 1° per B.t.u.
		Weight of the dry air, pounds	Weight of the vapor, pounds	Total weight of the mixture, pounds		
0	0.0383	0.08625	0.000069	0.08632	0.02082	48.04
10	0.0631	0.08433	0.000111	0.08444	0.02039	49.05
20	0.1030	0.08247	0.000177	0.08265	0.01998	50.05
30	0.1640	0.08063	0.000276	0.08091	0.01955	51.15
40	0.2477	0.07880	0.000409	0.07921	0.01921	52.06
50	0.3625	0.07694	0.000587	0.07753	0.01883	53.11
60	0.5220	0.07506	0.000829	0.07589	0.01852	54.00
70	0.7390	0.07310	0.001152	0.07425	0.01811	55.22
80	1.0290	0.07095	0.001576	0.07253	0.01788	55.93
90	1.4170	0.06881	0.002132	0.07094	0.01763	56.72
100	1.9260	0.06637	0.002848	0.06922	0.01737	57.57
110	2.5890	0.06367	0.003763	0.06743	0.01716	58.27
120	3.4380	0.06062	0.004914	0.06553	0.01696	58.96
130	4.5200	0.05716	0.006357	0.06352	0.01681	59.50
140	5.8800	0.05319	0.008140	0.06133	0.01669	59.92
150	7.5700	0.04864	0.010310	0.05894	0.01663	60.14
160	9.6500	0.04341	0.012956	0.05637	0.01664	60.10
170	12.2000	0.03735	0.016140	0.05349	0.01671	59.85
180	15.2900	0.03035	0.019940	0.05029	0.01682	59.45
190	19.0200	0.02227	0.024465	0.04674	0.01706	58.80
200	23.4700	0.01297	0.029780	.004275	0.01750	57.15

163. Specific Heat of Air.—The specific heat of a gas may be expressed in either of two ways: *i.e.*, the specific heat of constant

[1] From "Fan Engineering," Buffalo Forge Company.

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pressure, and the specific heat of constant volume. The reason for this has already been stated (Par. 6). In ventilating work the former quantity is the one involved. Its value as determined by Carrier is 0.2415 B.t.u.

Problems

1. Given wet-bulb temperature 66°, dry-bulb temperature 80°. Find dew point, per cent. saturation, and moisture content.

2. Given air at a temperature of 60° and containing 5 grains of water vapor per cubic foot. What is its relative humidity?

3. The air outside of a building is at a temperature of 31° and has a relative humidity of 84 per cent. On being drawn into the building it is heated to 70°. What is its relative humidity at the higher temperature?

4. Air at 80° is 87 per cent. saturated. When cooled to 55° what is its new moisture content?

5. Air at 25° has a humidity of 90 per cent. How much moisture must be added to give it a humidity of 50 per cent. when heated to 70°?

6. A room has a volume of 1800 cubic feet. The air is changed once per hour. The incoming air has a temperature of 35° and a relative humidity of 75 per cent. It is desired to maintain a humidity of 50 per cent. in the room, the temperature being 70°. How many gallons of water must be evaporated in 24 hours to do this?

CHAPTER XIII

VENTILATION

164. Ventilation Standards.—While the art of ventilating occupied rooms has advanced greatly during recent years, there are as yet no fixed standards as to what constitutes satisfactory ventilation. It is only very recently that many of the physiological effects of certain atmospheric conditions have been understood, and a satisfactory explanation of other phenomena is still lacking. The formulation of standard requirements has therefore been very difficult and further progress now depends upon their establishment rather than upon the mechanical problems involved in fulfilling them. The most agreeable atmosphere that we know of is undoubtedly that which exists outdoors on a sunny spring day; but the specific qualities which make it agreeable have not been definitely discovered. It is well known, however, that many other things beside the mere supplying of a sufficient quantity of air are necessary to provide comfortable conditions.

The effect of the atmospheric conditions upon the human body is twofold: namely, its effect upon the skin, and its effect when taken into the lungs. The former is largely a matter of removing heat from the body at the proper rate, while the latter is a question of supplying sufficient air of the proper cleanliness.

In the maintaining of what is now considered as satisfactory ventilation, the following factors must be taken into account:

1. Sufficient air supply, properly distributed.
2. Reduction of odors and impurities.
3. Removal of dust and bacteria to an acceptable amount.
4. Proper temperature.
5. Proper humidity.
6. Proper amount of air motion.

The first three of these factors concern the effect of the inhaled air, while the last three affect the rate of heat removal from the skin.

165. Sources of Air Pollution.—The percentage of oxygen absolutely necessary for human existence has been shown, in the preceding chapter, to be quite low, and a considerable reduction of oxygen may take place without even causing great

discomfort. In general, it may be stated that the quantity of air to be supplied for proper ventilation is governed by other factors which necessitate a greater quantity than that required to maintain a sufficient oxygen content.

The air of occupied rooms becomes the recipient of many polluting elements, the most important of which are the products of respiration. The average person breathes at the rate of about 17 respirations per minute while at rest. At each respiration, about 30½ cubic inches of air are inhaled or about 18 cubic feet per hour, which amounts to about 34 pounds of air in 24 hours or a little over 7 pounds of oxygen. The inhaled air loses about 5 per cent. of its oxygen content while in the lungs and gains from 3½ to 4 per cent. of carbon dioxide. The percentage composition of free air and of expired air, by volume, is about as follows:

	Free atmosphere, per cent. (approximately)	Expired air, per cent. (approximately)
Oxygen	20.9	15.4
Nitrogen	79.1	79.2
Carbon dioxide	0.03 to 0.04	4.03 to 4.04

In addition to carbon dioxide, water vapor is an important product of respiration. The moisture thus added to the air will increase the humidity above the comfort point unless the atmosphere is renewed with sufficient frequency.

There are also emanations from the mouth, lungs, and skin which give rise to disagreeable odors and which are believed by some to have a poisonous effect when taken into the lungs. Although this belief is not universally accepted, and although the exact effect of this organic matter is not known, common cleanliness alone demands that sufficient fresh air be supplied to dilute such impurities considerably.

There are other sources of air pollution, such as the products given off by the combustion in gas and oil lamps and from manufacturing processes. Gas lights give off carbon dioxide, water vapor, and traces of sulphuric acid. If the burners are not properly adjusted, carbon monoxide, which has a poisonous and sometimes a fatal effect, may also be generated. Table XXXVIII gives the amount of combustible consumed and the amount of carbon dioxide emitted per candlepower from gas lights.

TABLE XXXVIII.—AIR POLLUTION BY GAS LIGHTING

	Consumption of combustible per candlepower in cubic feet per hour	Carbon dioxide per candlepower in cubic feet per hour
Fishtail burner	0.802–0.527	0.494–0.304
Argand burner	0.0 –0.445	0.254
Welsbach burner	0.053–0.024	0.030–0.057

Manufacturing and chemical processes give off various gaseous impurities, but such conditions require individual study and no set rules can be given.

166. Amount of Air Required.—The proper amount of air supply has been determined from experience for different classes

TABLE XXXIX.—AIR SUPPLIED TO VARIOUS CLASSES OF BUILDINGS

	Cubic feet per hour per occupant	No. of renewals of air per hour
Churches, auditoriums and assembly rooms	1,200–1,800	
Theatres	1,000–1,200	
Grade schools	1,000–1,500	
High schools	1,200–1,800	
College class rooms	1,500–2,000	
Hospitals for ordinary diseases	2,500–3,500	
Hospitals for children	2,000–2,500	
Hospitals for contagious diseases	5,000–5,500	
Hospitals for wounded	3,500–5,000	
Barracks	1,000–1,800	
Living rooms in residences	1,200	1–2
Stairways and halls	600	½–1
Bedrooms	1,000	1½
Work shops	600–2,000	
Public waiting rooms	....	4
Public toilet rooms	....	10
Small convention halls	....	4
General offices	....	3
Private offices	....	4
Public dining rooms	....	4
Banquet halls	....	5
Basement restaurants	....	8–12
Hotel kitchens	....	4–6
Public libraries	....	3
Textile mills	....	4
Engine rooms	....	3–6
Boiler rooms	....	2–6
Railroad roundhouses	....	12

of buildings. For buildings such as theatres and schools, it is customary to provide a certain volume of air per minute for each occupant. For rooms where the number of occupants is variable or where there is pollution from sources other than respiration, sufficient fresh air is provided to renew that in the room a certain number of times per hour. For ordinary conditions of temperature and humidity, Table XXXIX gives the usual practice as to the amount supplied.

167. Methods of Measuring Air Supply.—When the air enters a room through but one or two ducts, the quantity can be directly measured by a pitot tube or anemometer, the use of which will be discussed in Chapter XV. Another method which in many cases is more convenient is based on the measurement of the carbon dioxide content of the air combined with our knowledge of the rate at which the carbon dioxide is added by the exhalation from the occupants.

Let V = volume of air admitted to the room in cubic feet per hour.
a = volume of CO_2 contained in a unit volume of the air admitted.
r_1 = amount of CO_2 per unit volume of air in the room at the beginning of the test.
r_2 = amount of CO_2 per unit volume of air in the room at the end of the test.
r = amount of CO_2 per unit volume of air in the room at any time during the test.
R = volume of room in cubic feet.
c = amount of CO_2 produced in the room, in cubic feet per hour.
t = time of experiment in hours.

During any small period of time dt, the amount of air entering the room is Vdt and the amount of CO_2 contained in the entering air is $aVdt$. The amount of CO_2 produced during the time dt is cdt. During the same interval, an equal volume Vdt leaves the room through the exhaust flues and its CO_2 content is $rVdt$. The net increase in the volume of CO_2 in the room is then

$$(aV + c)dt - rVdt = (aV - rV + c)\,dt$$

Let the increase in the CO_2 content of the air in the room per

cubic foot during the interval dt be represented by dr. Then the total net increase is Rdr. Equating the two

$$Rdr = (aV - rV + c)dt \tag{1}$$

and

$$dt = \frac{Rdr}{(aV + c) - Vr} \tag{2}$$

$$t = R\int_{r_1}^{r_2} \frac{dr}{aV + c - Vr}$$

$$t = R\left|_{r_1}^{r_2} \frac{1}{V} \log_e (aV + c - Vr)\right.$$

$$t = \frac{R}{V} \log_e \frac{Vr_1 - aV - c}{Vr_2 - aV - c}$$

$$V = 2.303\,\frac{R}{t} \log_{10} \frac{Vr_1 - aV - c}{Vr_2 - aV - c} \tag{3}$$

If $r_1 = r_2$, which means that there is no increase in the CO_2 content of the air in the room, then the amount entering the room, plus the amount produced must equal the amount leaving the room, or

$$aV + c = Vr_2$$

from which

$$V = \frac{c}{r_2 - a} \text{ and } r_2 = r_1 = a + \frac{c}{V} \tag{4}$$

If $c = 0$, then from (3) $V = 2.303\,\dfrac{R}{t}\log_{10}\dfrac{r_1 - a}{r_2 - a}$ (5)

Equation (4) is applied practically by assuming a certain production of CO_2 per hour per person, which figure is usually taken as 0.6 cubic foot. Equation (4) then becomes

$$C.F.H. = \frac{6000}{CO_2 - x} \tag{6}$$

in which

$C.F.H.$ = cubic feet of air per hour supplied to the room per occupant.

CO_2 = carbon dioxide content of the room air in parts per 10,000.

x = carbon dioxide content of the outside air in parts per 10,000.

This formula is recommended by Dr. E. V. Hill and is used by the Health Department of the City of Chicago. The chart in

Fig. 120 shows the air supply per person when any given CO_2 content exists in the room. The above method of determining the air supply does not apply when there is any source of carbon dioxide other than the lungs of the occupants.

168. Air Distribution.—Merely to supply enough air to a room is not sufficient for good ventilation; it must be distributed in a fairly uniform manner so that each occupant receives approximately the specified amount. The methods of distribution will be dealt with later. To determine the uniformity of distribution, the common method is to take measurements of the CO_2 content in different parts of the room and thus determine the variation of the quantity supplied per occupant at the different points from the average quantity.

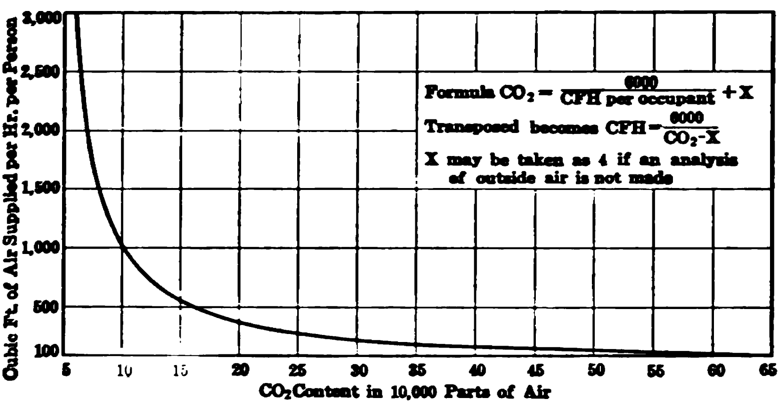

FIG. 120.—Chart showing air supply per person for various amounts of CO_2.

169. Temperature and Humidity.—One of the chief objects sought when air for ventilation is provided is the establishment of such conditions that heat will be removed from the human body at a rate which is favorable for comfort and health. Heat is lost from the body in three ways: by radiation, by convection, and by the evaporation of moisture from the skin. A relatively large amount of heat is lost by convection and consequently the temperature of the surrounding air and the amount of air movement are important factors.

The amount of heat lost due to the evaporation of perspiration from the skin depends upon the *relative humidity of the air* and upon the *amount of air motion.* It is also dependent, of course, upon the amount of perspiration which is given off by

the pores of the skin, more heat being lost by evaporation from the skin of a person who perspires freely than from a dry skin. Comfortable conditions can exist through a rather wide range of temperature and relative humidity *provided that the combination of the two is such as to cause the proper rate of heat loss from the body.* The air motion may also vary, but within rather narrow limits.

The chart in Fig. 121 showing the proper relation between the temperature and humidity was constructed by Dr. E. V. Hill from a series of tests made by Prof. J. W. Shepherd. From the center line of the "Comfort Zone" shown in the chart, it will be noted that equally comfortable conditions can be had with a

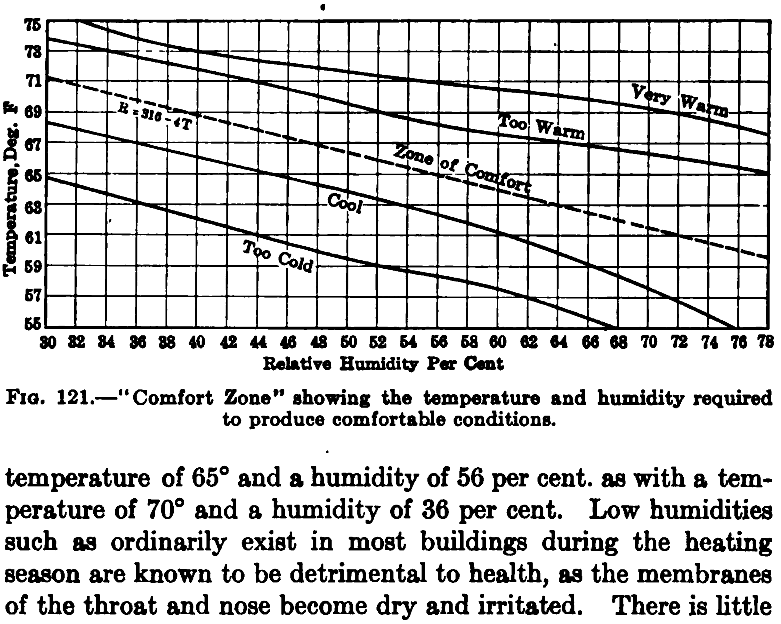

FIG. 121.—"Comfort Zone" showing the temperature and humidity required to produce comfortable conditions.

temperature of 65° and a humidity of 56 per cent. as with a temperature of 70° and a humidity of 36 per cent. Low humidities such as ordinarily exist in most buildings during the heating season are known to be detrimental to health, as the membranes of the throat and nose become dry and irritated. There is little doubt but that the proper humidification of the air of residences and other heated buildings is very beneficial from a physiological standpoint but there have been certain difficulties in the way of its universal application. Many of the devices intended for the purpose are entirely inadequate to supply the moisture required by even a moderate-sized room. There is also a general lack of appreciation of the quantities of moisture required, some idea of which was brought out in the preceding chapter. Another drawback is the tendency for moisture to be deposited on the

windows when even a moderate humidity is maintained in very cold weather. For these reasons, the application of artificial humidity has been limited to buildings of sufficient size and of such a character as to make practicable the use of an air washer or other rather elaborate means for humidification.

Excessive humidity, on the other hand, is undesirable, as it causes a feeling of intense discomfort, especially when accompanied by high temperature, because of its effect in lowering the rate of evaporation from the skin and therefore retarding the process of heat removal from the body. According to Prof. Foster, about 4 pounds of water are given off by an adult man under extreme conditions in 24 hours, of which 2½ pounds are evaporated from the skin and the remainder is contained in the expired air. Under average conditions, the amount given off is about one-half the above. The heat given off from the body will vary from about 335 to 460 B.t.u. per hour depending upon the age, sex, diet, amount of exercise, etc. About 15 B.t.u. of this amount are given off with the expired air and 35 B.t.u. are contained in the moisture with which the expired air is saturated. Approximately 70 B.t.u. are contained in the moisture which is evaporated from the skin, leaving about 250 B.t.u. to be lost by convection and radiation. The last two quantities especially are extremely variable under different atmospheric conditions. In a hot, dry atmosphere, for example, the air temperature may be higher than that of the body and no heat can be given off by convection or radiation. The evaporation of perspiration from the skin must then be depended upon to remove all of the excess heat from the body.

In crowded rooms, the heat and moisture given off from the bodies and from the exhalations of the occupants may render the atmosphere extremely uncomfortable, so that cooling and dehumidifying are required. It has been demonstrated repeatedly that the depressing effect of a so-called stuffy atmosphere is due to its action on the skin as much as or even more than to its effect on the lungs.

When the air is artificially cooled, it has been found that the inside temperature must be raised somewhat as the outside temperature increases, so that the shock to the sensations of one entering from the outside will be minimized. The inside temperature should not be more than 10° or 12° below the outside, as a maximum.

170. Air Movement.—The rate of evaporation of moisture from the skin depends, in addition to the temperature and humidity of the atmosphere of the room, upon the continuous renewal of the aerial envelope surrounding the body. Unless this renewal takes place, the air immediately surrounding the body increases in temperature and moisture content to such an extent that the skin evaporation is retarded to an uncomfortable degree. A proper circulation of the air within a room also prevents the immediate reinspiration of the air expired from the lungs and diffuses it throughout the room. The motion of the air, however, should never be such as to cause uncomfortable drafts. In general, a movement toward the face is greatly preferable to one from the rear. An air movement of about 2 feet per second is considered to be permissible, but a much greater velocity is uncomfortable. Air movement may be directly measured by injecting into the air clouds of smoke and timing their movement across the room. Toy balloons are also used for this purpose.

Cubic space is an important factor in ventilation, particularly in crowded rooms, for with too small a volume per person it may be impossible to move the required amount of air through the room without giving rise to unpleasant drafts. Dr. Billings recommends the following as the minimum amount of space to be allowed per occupant.

Lodging or tenement house..	300 cubic feet per person
School room...............	250 cubic feet per person
Hospital ward.............	1,000–1,400 cubic feet per person
Auditoriums...............	200 cubic feet per person

In computing the cubic space for this purpose, all space over 12 feet above the floor should be neglected.

171. Odors.—Another function of ventilation is the removal or reduction of odors, the most common and most objectionable of which arise from the bodies of the occupants. The sources of these odors are emanations from the throat and lungs, the perspiration from the skin, and soiled clothing. In factories, odors are created by industrial operations of various sorts.

The so-called "crowd smell" is not harmful of itself, for it has been shown that healthful existence is quite possible in such an atmosphere. Repulsive odors are indirectly harmful, however, in that they cause the occupants of the room to breathe

less deeply; but regardless of their actual physiological effect, modern standards of cleanliness require that sufficient air be supplied to occupied rooms to maintain a wholesome atmosphere.

As yet, no satisfactory standard has been found for the measurement of odors.

172. Ozone.—Ozone is used to some extent as a means for counteracting odors and bacteria. Ozone is simply a form of oxygen in which the molecule consists of three instead of two atoms. The additional atom is readily liberated and the substance is consequently an active oxidizing agent. Ozone is present in very minute amounts in the atmosphere.

When injected into the atmosphere of a room with a concentration of not more than 1 part per million, ozone is capable of obliterating even very marked odors. The exact action which takes place is at present a matter of debate. By some it is believed that ozone actually destroys the odors through its oxidizing action. It is known, however, that it is quite possible to compensate one odor with another so that its effect upon the olfactory membrane is neutralized, and it may be that the real action of the ozone is a masking of the odors by what is called "olfactory compensation" rather than a destroying of them.

It is very essential that the concentration of the ozone be kept very low, for in an atmosphere of more than about 1 part per million of ozone, serious irritation of the throat and lungs is liable to result.

The common method of producing ozone is by means of an electrical discharge at high voltage. Several commercial machines are available for the purpose.

173. Dust and Bacteria.—The air, especially that of cities, contains a large amount of dust in very finely divided particles. These particles consist of many different substances, most of which are mineral. In large cities, tons of cinders and smoke particles are cast out into the air annually, which adds to the production of dust from other sources. Dust in itself is not particularly injurious to health but it serves as a carrying medium for all sorts of bacteria.

Several methods of determining the dust content of air have been devised. The most successful scheme is to draw a sample of air into a suitable cylinder containing a glass disc coated with an adhesive varnish and so placed that the indrawn air impinges upon it. The number of dust particles determined by microscopic

count affords an indication of the amount of dust in the air. Dust can be quite thoroughly removed from air by means of the air washer, to be described later.

174. Methods of Introducing Air.—In providing ventilation for a room, it is necessary to adopt a definite scheme for the introduction of fresh air and the removal of the vitiated air. When the air quantities are small the leakage around the windows may be relied upon as a means for permitting the escape of the air, but in general, it is necessary to install a system of vent flues.

There are two general methods of circulating the air through a room. In the upward system, the air is introduced through the floor or through the side walls near the floor and is removed near the

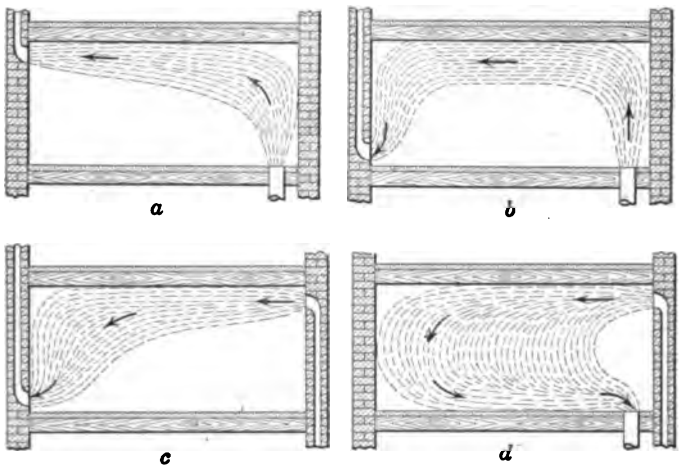

Fig. 122.—Effect of various locations of inlet and outlet.

ceiling. In the downward system, the air is introduced through registers in the side walls located from 7 to 10 feet above the floor and is removed near the floor. The former method is especially adapted to theatres and auditoriums where a large number of small openings can be provided in the floor, thus securing a very even distribution. The upward system is also suitable for restaurants and rooms where there is smoking or where other impurities or odors are created which have a natural tendency to rise. The downward system is used in schools, hospitals, etc. where the occupants are not many and where it is not practicable to have openings in the floor.

The relative location of the inlet and outlet openings greatly

affects the thoroughness of the air renewal throughout the room. It has been demonstrated that the most effective scheme is to place the outlet near the floor and on the same side of the room as the inlet. The effects of various locations of the inlet and outlet are shown in Fig. 122.

Problems

1. A test made in a room in which there are several people shows a CO_2 content of 12 parts per 10,000. What quantity of air is being supplied per hour per occupant?

2. A test of the air of an occupied room shows a CO_2 content of 13 parts per 10,000. Outside air contains 3½ parts per 10,000. How much air is being supplied per hour per occupant?

CHAPTER XIV

HOT-AIR FURNACE HEATING

175. Furnace Systems.—The hot-air furnace is widely used throughout the country in the heating of residences of moderate size. In addition to its simplicity and relatively low cost, it has the great advantage of providing fresh air for ventilation. It is especially well adapted to moderate climates where, for a large part of the winter, heat is needed only in the morning and evening.

As brought out in Chapter III, the greatest disadvantage in a furnace system is the fact that the force producing circulation, being dependent upon the relatively slight difference in density between the heated and unheated air, is quite small and is often insufficient to overcome the resistance of the piping or the pressure of a very strong wind blowing against the building. These difficulties can often be overcome, however, by intelligent design of the system. The size of the building which a hot-air furnace can serve is limited because of the friction in the horizontal piping. The practical limitation to the length of horizontal ducts which can convey the required volumes of air is about 20 feet. It is sometimes feasible to install two separate furnaces and thus avoid pipes of excessive length.

There are many forms of furnaces on the market, some of which are of indifferent design and workmanship. The non-success of many furnace installations is usually due to this fact and to the lack of intelligent planning of the piping system. A common mistake, brought about by the endeavor to minimize the cost of the installation, is the use of a furnace of insufficient size. As a result, a very hot fire must be maintained and the flue gases leave the furnace at a high temperature, necessitating the use of an excessive amount of fuel.

In a furnace system the heat is absorbed by the air as it passes through the furnace and is carried by the air to the rooms above. The air circulates through the rooms, giving up heat to the objects in the room and to the walls. The walls and the con-

tents of the room remain at a slightly lower temperature than the air. If no foul-air flues are provided, the air entering the room must eventually find its way out through the cracks around the windows, through the walls themselves, or through the doors into other rooms; for otherwise the flow of air into the room could not be maintained. It sometimes happens, in a tightly constructed building, that this leakage is insufficient, and the flow of air to the room is retarded. A foul-air flue from each room is therefore desirable, although in the average residence its initial cost is seldom deemed warranted.

The air entering the furnace may come either from outside or may be partially or wholly re-circulated from the house. It is best to provide both means of air supply, so that either may be used as conditions demand. When only a few people are in the building, the air may be re-circulated, but for a number of people it is very desirable that an ample supply of fresh air be introduced. The economy of a hot-air system is affected by the proportion of the air taken from each of these sources. When all the air is re-circulated from the house, then the economy of the hot-air furnace is about the same as that of a steam or hot-water plant; but when air is taken from outside, then a certain amount of heat is used in warming it up to the temperature of the room, and this heat is not available for supplying the heat losses from the building. But heat used in this way should be considered as the price of ventilation and should not be charged against the efficiency of the system.

176. Furnaces.—The hot-air furnace consists fundamentally of a firepot and a series of passages for the flue gases, surrounded by a metal or brick casing. The air circulates through the space between the furnace proper and the casing, absorbing heat from the hot surfaces of the firepot and gas passages. The gas passages are usually formed by a simple casting called a "radiator."

Hot-air furnaces are quite varied in design. In general there are two types: those with the radiator at the top of the furnace, as in Fig. 123; and those with the radiator near the bottom of the furnace, as in Fig. 124. Occasionally, in cheap furnaces, the radiator is left off entirely. For the best possible efficiency in any furnace the entering air should first come into contact with the surfaces behind which are the coldest flue gases and the air leaving the furnace should pass over the hottest surfaces. This ideal condition is difficult of realization, for mechanical

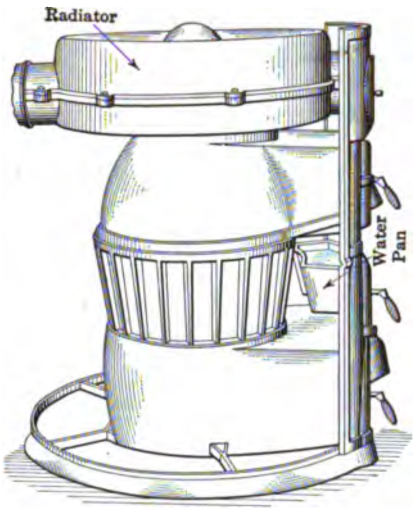

FIG. 123.—Furnace with radiator at the top (casing removed).

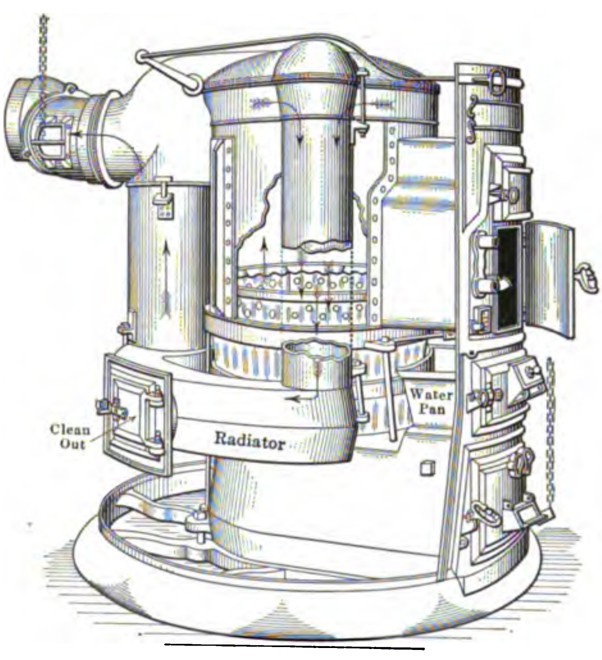

FIG. 124.—Furnace with radiator near bottom (casing removed).

reasons, but the furnace which most nearly approaches it will in general be the most efficient.

The heating surfaces of a furnace may be divided into two classes: (a) direct heating surfaces, which are those which are in contact with the fire or which receive heat by direct radiation from the fire; and (b) indirect heating surfaces, which are heated only by the hot gases. In addition there are some surfaces which receive heat only by conduction from the heating surfaces proper, such as projections and braces, these being called "extended" surfaces. The parts of such surfaces which are more than about 2 inches from actual heating surface are of doubtful effectiveness, however.

All of the heating surfaces give up heat to the air entirely by convection. The amount of heat transmitted through the heating surfaces of course increases as the difference in temperature between the air and the products of combustion increases. The effectiveness of the heating surfaces decreases as the distance from the fire increases and direct heating surfaces are naturally more effective than indirect heating surfaces. The more rapid the flow of air over the heating surfaces, the greater will be the amount of heat removed.

Since the effectiveness of the heating surfaces depends upon the design of the furnace, it is impossible to base the capacity of the furnace upon the amount of heating surface. Roughly, however, the heat transmission may be assumed to be, on an average, from 1000 to 1500 B.t.u. per hour per square foot of surface.

177. Furnace Construction.—The firepot and radiator are usually made of cast iron, although steel is sometimes used. There is no appreciable difference in the thermal conductivity of the two materials. It is essential that the joints between the different castings be air-tight so that the products of combustion cannot escape and be carried to the rooms above. The joints, therefore, are of a modified tongue and groove design, the grooves being filled with a special cement and the castings drawn and held together with draw bolts. Joints should be as few as possible and vertical joints should be avoided.

The firepot is usually slightly conical and should be deep enough to contain sufficient coal to last for 8 hours, leaving enough unburned coal on the grates at the end of that time to ignite the fresh charge of fuel. With hard coal this means that

the depth should be sufficient to allow for 50 pounds of coal being placed on the grate per square foot of grate. Coke or soft coal will require a greater depth of firebox than anthracite coal. The grate area should be from 1:25 to 1:12 of the area of the heating surface, the proportion depending upon the kind of fuel and the size of the furnace—the larger the furnace, the smaller the ratio. If anthracite coal is used the ratio should not exceed 1:25. If bituminous coal is used it should be 1:20 and for coke about 1:15 for furnaces of average size.

For burning soft coal some furnaces are provided with an auxiliary air supply so arranged that heated air is introduced into the firepot above the fuel bed, mixing with the combustible gases and promoting complete and smokeless combustion.

The furnace casing is usually of galvanized iron, although large furnaces are sometimes enclosed by brickwork. When a galvanized-iron casing is used, insulation is obtained by providing an inner casing of black iron or tin with an air space between the inner casing and the outer casing of about 1 inch. The area between the furnace and casing should be sufficient so that no appreciable resistance is interposed to the circulation of air through the furnace. In small furnaces the velocity should not exceed 250 feet per minute and in larger furnaces 300 to 250 feet per minute. These figures apply only to gravity circulation.

Furnaces are rated by the manufacturers either upon a basis of the volume of the building to be heated or upon the total cross-sectional area of the warm air ducts. Inasmuch as these ratings usually represent about the maximum capacity of the furnace, it is well to choose a furnace of 25 to 50 per cent. excess capacity.

178. Humidification.—The hot-air furnace system affords a particularly favorable opportunity for humidification, but unfortunately most of the manufacturers have been extremely backward in providing adequate apparatus for adding the necessary amounts of moisture to the air. Most furnaces are equipped with some sort of a "water pan" which is usually installed near the bottom of the furnace. This location is entirely wrong, for the air as it enters at the bottom of the furnace has the least capacity for absorbing moisture. To be effective, the humidifying apparatus should be placed where the hottest air will pass over it, *i.e.*, at the furnace outlet. Few realize that in order to maintain a proper humidity in even a small house there must be evaporated

hourly a quantity of water of the order of 10 pounds. To be satisfactory, the water pan must therefore be kept filled automatically from the water-supply system. Fig. 125 shows a humidifier which is located at the top of the furnace and is automatically filled. The amount of water evaporated increases with the amount of air passing through the furnace and with the temperature of the air, making the apparatus to some extent self-regulating. Accurate automatic regulation is impossible, however, without a system of humidity control such as will be described later.

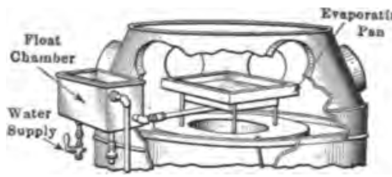

FIG. 125.—Humidifier.

179. Cold-air Pipe.—The air supply to the furnace may be taken from outside or can be re-circulated from the house. It is also quite feasible to take only a certain amount of air from outside and to supply the remainder by re-circulation. With complete re-circulation the advantage of ventilation is entirely lost but the system is somewhat more economical. The cold-air duct may be of galvanized iron or may be constructed of tile and placed beneath the basement floor. It should come from the side of the house which is subject to the prevailing winds. It is sometimes desirable to have cold-air ducts leading to different sides of the house so that the supply of cold air may be taken from the windiest side. The cross-section of the cold-air duct should be 80 per cent. of the aggregate area of the supply ducts leaving the furnace.

The re-circulating duct should be brought from the coldest part of the house and from some room such as the hall which has other rooms leading into it. The side of the stairway, the lower stairway risers, or the space in front of large windows are good locations for the re-circulating register.

If it is desired to re-circulate partially and to take the balance of the air from outside, the re-circulating pipe should be brought to the furnace independently of the fresh-air pipe, and a deflecting plate placed in the air space under the furnace. If this is not done, the air may come in from the outside and pass up

the re-circulating pipe instead of through the furnace. Both the fresh-air pipe and the re-circulating pipe must be provided with dampers.

It is a common error to make the re-circulating pipe of a furnace system too small. The area of the re-circulating pipe should be not less than three-fourths the combined area of the hot-air pipes, and it is better to have it equal to their combined area.

180. Hot-air Pipes.—The furnace should be centrally located, or if the coldest winds come from a certain direction, it can be located toward that side of the house. The pipes leading from the furnace should be as short and direct as possible; long horizontal pipes should be avoided.

The horizontal pipes are called **leaders**; the vertical pipes **flues** or **risers**. Leaders are usually made of round pipe. All leaders should be given the same pitch of at least 1 inch per foot and should leave the furnace at the same elevation. They should be insulated with asbestos paper, or if extending through a very cold part of the basement, with an air-cell covering. It is desirable to provide a damper in each pipe so that the distribution of the air among the different rooms can be adjusted. The risers should in every case be installed in an inside partition, as the cooling effect, when placed in an outside wall, would greatly retard circulation, besides causing an excessive waste of heat. It is usually necessary to limit the depth of the riser to 4 inches, so that it may be enclosed in the studding. The width also is sometimes limited by the distance between the studding and many furnace systems suffer from this source. It is sometimes necessary to run two risers to large second-floor rooms when space is not available for a single riser of sufficient size. Architects often fail to realize the importance of providing sufficient space for this purpose.

Risers, when made of single-walled pipe must be insulated with asbestos paper to protect the woodwork and a clearance on all sides of at least ¼ inch must be left. Double-walled pipe which has an air space between the walls is becoming widely used. The air space serves as an insulator and greatly decreases any possible fire hazard as well as reducing heat loss from the pipe. When double-walled pipe is used the proper size should be selected so that the net inside area will not be reduced below the computed requirements. Bright tin is ordinarily used for all piping.

The leader is connected to the riser by means of a fitting called a "boot" shown in Figs. 126*a* and 126*b*. The form shown in Fig. 126*a* is preferable as it interposes less resistance to the flow of air.

The air is delivered into the room through a register of the form shown in Figs. 127 and 128. Floor registers have the advantage that they may be made of any size and may be placed in any part of the room. They are often favored because the air

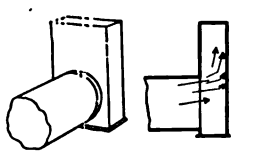

Fig. 126*a*.—Boot of improper design.

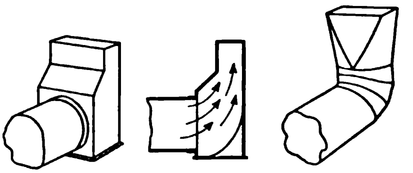

Fig. 126*b*.—Boots of good design.

leaving them does not deposit dust on the walls as does the side-wall register. Floor registers, however, are very insanitary as they collect great quantities of dirt; and they also frequently necessitate cutting the carpets or rugs. On the whole, the side-wall register is much to be preferred. Registers are provided with means of cutting off the flow of air in the form of louvres or,

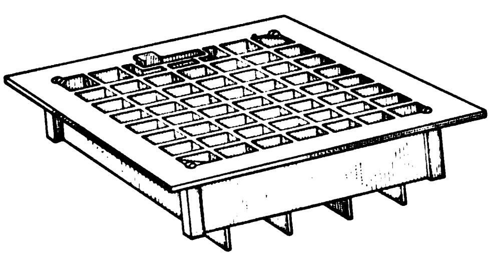

Fig. 127.—Floor register.

Fig. 128.—Wall register.

in the side-wall type, a single shutter of sheet metal. The shutters in some of the registers should be omitted, so that by no possible chance could all of the air supply be cut off; for with no air circulating through the furnace, the danger of overheating and burning out the firepot is great. It is often convenient to supply a first-floor register and a riser from a single leader. This can be very satisfactorily accomplished by means of the arrangement shown in Fig. 129. The free area of an ordinary register

is only about half of its gross area and its size must therefore be about double that of the pipe which supplies it. For a floor

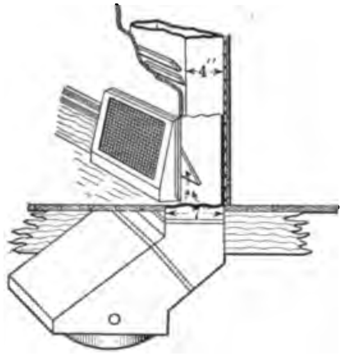

Fig. 129.—Method of connecting first floor register and riser to a single leader.

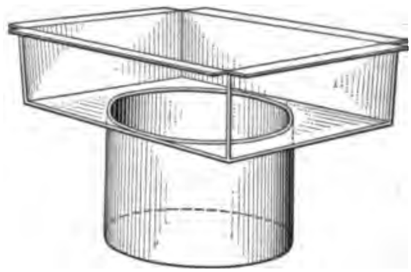

Fig. 130.—Box for floor register.

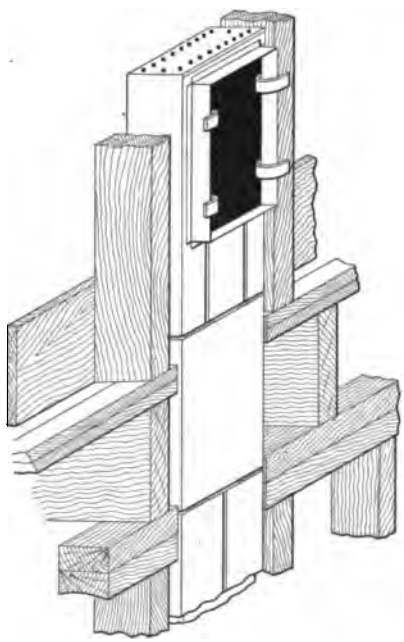

Fig. 131.—Stack and register frame—double walled pipe.

register a box of the form shown in Fig. 130 is provided and for a wall register a frame of the form shown in Fig. 131 is used.

Fig. 132 shows the general arrangement of a furnace system.

181. Size of Hot-air Pipes.—There are two methods of

figuring the size of the hot-air pipes, the B.t.u. method and the volume method. The former is the more rational and is the one recommended.

Example of B.t.u. Method.—Assume that the heat loss from the given room is 12,000 B.t.u. per hour and that the air enters at 140°, room temperature being 70°. Each cubic foot of air entering the room will give up enough heat to lower its temperature from 140° to 70°. The amount of heat given up when a cubic foot of air is cooled 1° is approximately $\frac{1}{55}$ B.t.u.

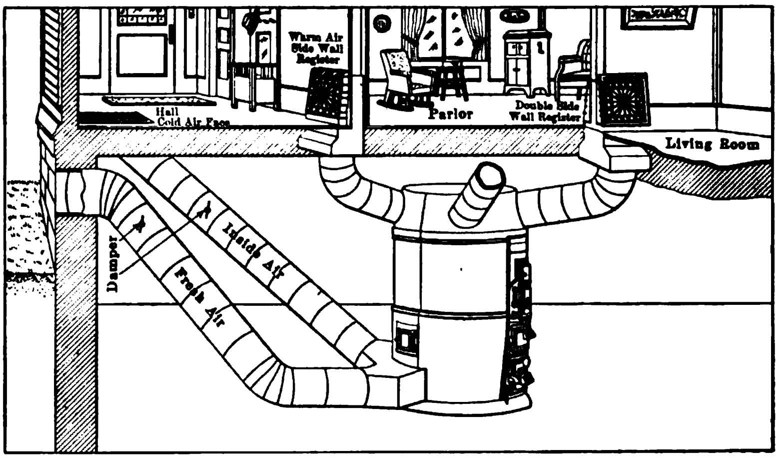

FIG. 132.—General arrangement of furnace system.

Therefore, the heat given up per cubic foot is $\frac{140-70}{55} = 1.27$ B.t.u. The volume of air required per hour $= 12{,}000 \div 1.27 = 9460$ cubic feet. Allowing a velocity of 250 feet per minute in the hot-air pipe, the required area of the hot-air pipe is $\frac{9460}{250 \times 60}$ $= 0.63$ square foot.

The velocity of air for first-floor leaders may be taken as 3 to 4 feet per second, for second-floor leaders 4 to 5 feet per second, and for third-floor leaders 5 to 6 feet per second. The risers leading to second- and third-floor rooms may have a velocity as high as 400 feet per minute.

Registers should be proportioned so as to give a velocity of 2 to 3 feet per second on the first floor and 3 to 4 feet per second on the floors above, on the basis of the effective area of the register.

Volume Method.—In the volume method the area of the hot-

air pipe is assumed to be purely a function of the size of the room, no account being taken of the heat losses. This method is manifestly inaccurate as the amount of air required depends of course upon the heat lost from the room. For rooms of average proportions and of ordinary construction, the volume method is usually successful, however, if carefully applied. The chart in Fig. 133 gives the size of leaders and risers required for rooms of various dimensions. It is permissible to reduce the size of the leader to which a riser is connected, as indicated by the chart, because of the acceleration of the circulation due to the stack effect of the riser.

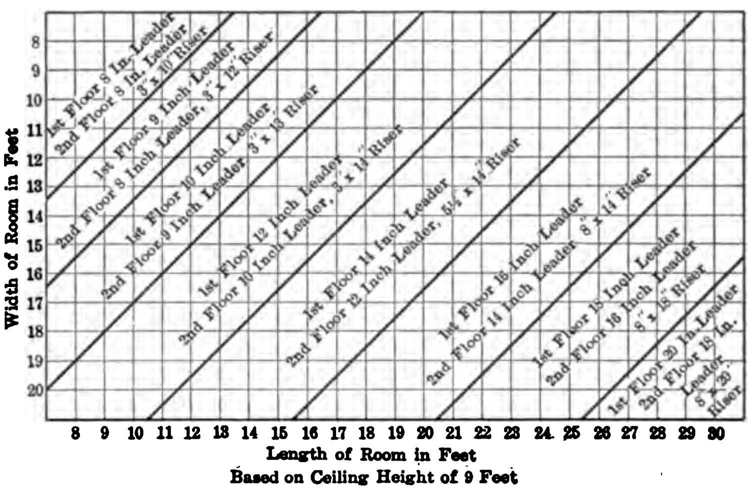

FIG. 133.—Size of hot air pipes for rooms of various dimensions.

182. Suggestions for Hot-air Piping.—The following rules should be observed in the installation of the leaders and risers.

Never use smaller than 8-inch pipe for the leaders.

When a leader is more than 15 feet long, add 1 inch to the diameter for each 4 feet or fraction thereof over 15 feet and increase the riser to correspond.

Rooms on the sides of the house exposed to prevailing winds should have one size larger pipe than rooms of equal size on the other sides of the house. If the exposed rooms have a considerable amount of glass surface, they should have pipes two sizes larger.

Avoid horizontal pipes under the second floor if possible.

When unavoidable, make them one-fourth larger than the risers and give them all the pitch possible, avoiding square angles.

In Table XL are given the equivalent areas of round pipes, rectangular pipes, and registers.

TABLE XL.—EQUIVALENT SIZES OF PIPES AND REGISTERS[1]

Diameter of round pipe	Area of pipe, square inches	Size flat riser pipe	Size side-wall register	Size round floor register	Size rect. floor register
8	50	3½×14	8×12	12	8×12
9	64	4×16	10×12	14	10×12
10	78	4×20	12×12	14	10×16
11	95	6×16	12×15	16	12×15
12	113	6×19	14×15	18	12×20
13	132	6×22	14×18	18	14×18
14	154	8×19	16×18	20	14×22
15	176	8×22	16×20	24	16×20
16	201	8×25	18×20	24	16×24
17	227	10×23	18×24	24	18×24
18	254	10×26	20×24	24	18×27
19	283	12×24	20×26	28	20×26
20	314	12×26	22×26	28	20×30
21	346	12×29	24×27	30	22×30
22	380	14×27	24×30	30	24×30
23	415	14×30	27×27	30	24×32
24	452	14×32	28×28	36	24×36

All measurements in inches.

The circulation to a room which is unfavorably situated or which has a considerable amount of glass surface may be aided by installing a re-circulating duct from a register located beneath the windows to the lower part of the furnace casing.

183. Foul-air Flues.—It is important that means be provided for allowing the escape of air from the various rooms; for fresh warm air cannot enter unless it can displace an equal volume of room air. A fireplace is a very good form of foul-air flue and if the house is well provided with fireplaces, no other foul-air flues are necessary. Where several rooms open into each other and one of them has a fireplace, this may serve as a foul-air flue for all the rooms.

The cracks around the windows and doors often serve to allow air to escape, but when located on the exposed side of the house, the pressure of the wind prevents the outflow of air and the air

[1] From "Handbook of National Warm Air Heating and Ventilating Association."

supply to the room may be greatly retarded. For such rooms it is well to provide either a re-circulating duct or a foul-air flue.

The foul-air flues should be enclosed in the inside partitions and the registers should be placed at the baseboard. The reason for such an arrangement is that the hot air entering the room opposite the window surfaces rises to the ceiling and passes along the ceiling to the windows where it is cooled, dropping to the floor and passing along it to the foul-air register. The hot-air register should be a sufficient distance from the foul-air register so that the hot air will not pass directly to the foul-air flue.

A cheap foul-air flue can be made by having a register in the baseboard opening into the space between the studding, selecting a space that is open to the attic. A ventilator placed on the roof discharges the air from the attic. No two rooms should be connected to the same studding space. A still better arrangement is to extend each flue separately to the ventilator.

The area of the foul-air flues should be at least 80 per cent. of that of the warm-air flues and they are often made equal in area to the latter.

184. Forced Circulation.—Furnace systems are sometimes installed in which the circulation is produced by a fan. The principal advantage of such an arrangement is that the circulation is positive and is not affected by weather conditions. The fan, usually of the disc or propeller type, is placed in the cold-air inlet to the furnace and forces the air through the furnace and thence through the hot-air pipes to the rooms. Furnace systems with forced circulation are used principally where a considerable amount of air is required for ventilation and where an outfit is desired of lower first cost than an ordinary fan system.

185. Pipeless Furnaces.—One type of furnace which deserves but brief mention is the so-called "pipeless" furnace system. In this system a single register is used, located immediately above the furnace, and consisting of two sections, one section supplying hot air and the other section being connected to a re-circulating duct leading back to the base of the furnace. It is evident that with such an arrangement the room above the furnace will receive the greatest amount of heat and that all the other rooms can receive heat only by the natural circulation of air through them. The advantage of the pipeless furnace is its low cost. It is strictly limited to very small houses or bungalows and is not successful if installed outside

of this field. It lacks one of the chief advantages of the ordinary hot-air system—the providing of fresh air for ventilation.

186. Test of Hot-air Furnace.—The following is a summary of the results of a heat analysis of a hot-air furnace made at the University of Michigan.[1]

Test No.	7	11
2 Length of test—hours	30.00	31.00
3 Number of firings	2.00	4.00
5 Inlet temperature of air	50.60	39.60
6 Average temperature of heated air	113.70	109.20
7 Temp. of wet-bulb thermometer	70.80	64.70
8 Temperature of dry-bulb thermometer	112.00	107.70
9 Humidity, per cent	11.00	7.00
12 Volume of air delivered, cubic feet per minute	1,110.00	1,284.00
14 Temperature of gases over fire, deg. F	....	691.00
15 Temperature of gases in breeching, deg. F	309.00	318.40
16 Draft in breeching, inches of water	0.07	0.076
17 CO_2 content of flue gases, per cent	10.26	8.10
21 Kind of fuel	Mixed stove and egg anthracite	Gas coke
22 Total weight of fuel fired	255.00	330.50
23 Total weight of ash and refuse	37.00	16.50
24 Proximate analysis of fuel, per cent.		
Moisture	0.78	6.00
Volatile	4.75	3.60
Fixed carbon	88.61	86.10
Ash	12.86	4.30
26 Heat value per pound as fired	12,856.00	13,026.00
28 Total water evaporated from water pan, pounds	62.00	123.00
Heat balance, per cent.		
43 Heat input in fuel	100.00	100.00
44 Heat absorbed by air	61.60	63.00
45 Heat given to water	2.05	3.10
46 Heat given to air, gross	63.65	66.10
47 Heat lost up the stack	11.65	13.50
48 Heat lost in unburned fuel	1.60	0.70
49 Heat lost from furnace by radiation	11.00	8.83
50 Unaccounted for efficiency	12.10	10.87
51 Efficiency—net (Item 46) per cent	63.65	66.10
52 Efficiency—gross (Items 46 + 49 + ½ of 50) per cent	80.70	80.36

[1] "Heat Analysis of a Hot-air Furnace," by JOHN R. ALLEN, *Trans.* A. S. H. & V. E., 1916.

It will be noted that the heat given up to the air passing through the furnace is from 63 to 66 per cent. of the heat input in the fuel. In most installations, however, the heat radiated from the furnace is largely utilized, making the "gross" efficiency about 80 per cent.

Problem

1. Compute the required size of the hot-air pipes and of wall registers for the following rooms.

Room No.	Heat loss from room	Floor
1	16,000	First
2	10,800	Second
3	8,700	Third
4	5,000	Second

CHAPTER XV

DESIGN OF FAN SYSTEMS

187. Types of Fan Systems.—Fan systems are installed primarily to provide fresh air for ventilation, although in some classes of buildings they are preferable from a heating standpoint also. There are two general types of systems in use. In a simple ventilating system the heat lost through the walls of the building is supplied by direct radiation, and the air for ventilating is introduced at a temperature but slightly above that of the rooms. In the second type the heat losses are taken care of by the fan system and no direct radiation is installed. This means that the air must be heated considerably above the room temperature, to a point dependent upon the heating requirements.

The former system is usually the more suitable for buildings which require ventilation only part of the time, such as schools and office buildings. The proper temperature can be maintained in the building by means of the direct radiation and the fan system need be operated only when ventilation is required. Furthermore, the amount of air introduced can be limited to that actually required for ventilation. The fuel consumption is thus reduced to a minimum if the system is carefully operated. An objection to the combination system is the high first cost, which is considerably above that of a fan system alone.

The second type, in which the heat losses from the building are supplied by the fan system, is most suitable for buildings which must be continuously ventilated. A system of this type must be operated whenever the building is to be heated and much more air may be introduced than is required for ventilation. In theatres and churches, which need only be heated when ventilation is required, this arrangement is suitable. In some cases means can be provided for re-circulating the air from the building when desired instead of drawing in fresh air. The fresh air need then be introduced only when ventilation is needed, the air being re-circulated when the building is unoccupied. Such fan systems, supplying both the heating and ventilating

requirements, are often used in industrial buildings and are termed "hot-blast" systems.

188. General Arrangement.—The usual arrangement of a simple fan system for heating is shown in Fig. 134. The air is drawn from the outside over a set of tempering coils which heat it up to about 70°. It is then drawn through a set of heating coils and discharged by the fan into the system of supply ducts. In the heating coils its temperature is raised from 70° to whatever temperature is demanded by the heating requirements. A bypass damper is provided beneath the heating coils by means of which tempered air may be admitted to the duct system and

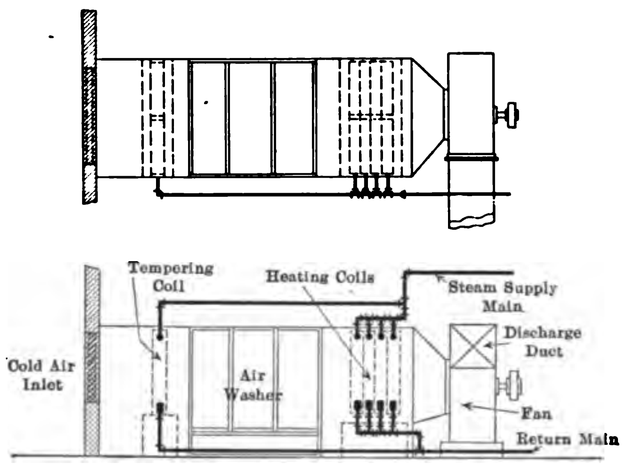

FIG. 134.—General arrangement of fan system.

the temperature of the air in the ducts may be regulated to suit the heating requirements.

Fan systems may be arranged either as "draw-through" or "blow-through" systems. In the former the heating coils are located at the fan inlet and the fan discharges directly into the duct system. Such an arrangement is slightly the more efficient but the blow-through system is on the whole the more suitable for most classes of buildings as it permits of a better arrangement of the mixing dampers.

189. Calculation of Air Quantities.—When ventilation only is considered the quantity of air to be handled by the fan is governed by the number of people in the building and the amount of air to be supplied per person. In Chapter XIII the con-

siderations affecting ventilation requirements were discussed, and in Table XXXIX, page 181, are given the quantities required per person or the number of air changes per hour for various classes of buildings.

If Q is the total quantity of air to be introduced per hour and H_v is the heat which must be added to the air in B.t.u. per hour, then:

$$H_v = QD_2C_p(t_2 - t_1) \quad (1)$$

in which C_p = specific heat of air at constant pressure (=0.2415).
t_1 = temperature of outside air.
t_2 = temperature of inside air.
D_2 = density of air at temperature t_2 in pounds per cubic foot.

In this expression the heat absorbed by the water vapor is neglected but the formula is sufficiently accurate for ordinary purposes. If the minimum outside temperature, for which the system is to be designed, is 0° and the inside temperature is 70°, then $D_2 = 0.07495$ and formula (1) becomes

$$H_v = Q \times 0.07495 \times 0.2415(70 - 0)$$
$$H_v = 1.27Q \quad (2)$$

In the case of a fan system supplying the heat which is lost through the wall and glass surface, this amount of heat must be added to the air delivered.

The air after entering the rooms is cooled to room temperature and discharged to the outside at that temperature. The total heat added to the air may therefore be thought of as being divided into two parts: (*a*) that which would be added were ventilation only being considered, which is the quantity required to raise the air from the outside temperature to room temperature, and (*b*) the additional amount added to supply the heat lost through the walls. The latter quantity may be expressed in the form

$$H_h = QD_2C_p(t_3 - t_2) \quad (3)$$

in which t_3 = temperature at which the air is delivered.
D_2 = density at room temperature, pounds per cubic foot.

The air volume Q is ordinarily taken at room temperature, assumed to be 70°.

Then

$$H = H_v + H_h = QD_2C_p(t_2 - t_1) + QD_2C_p(t_3 - t_2) \quad (4)$$

The quantity of air Q may be governed either by the ventilating requirements or by the heating requirements. If the heat loss from the building is large, a large quantity of air at the maximum temperature to which it is practicable to heat it, must be introduced, and this quantity may be greatly in excess of that required for ventilation. On the other hand, if the room is to contain a large number of people and if the heat loss is comparatively small, then the quantity of air will be fixed by the ventilation requirements and the temperature of delivery, t_3, will be fixed by the heating requirements.

Example.—Consider an auditorium which seats 400 people and which is to be ventilated with an allowance of 1500 cubic feet per hour per person. Assume that the fan system is to supply the heat losses as well as the ventilation requirements, and that a temperature of 68° is to be maintained. Let H_h, the heat loss through the exposed wall and glass surface be 860,000 B.t.u. per hour, and assume that the air is to be delivered, under maximum conditions, at a temperature of 140°. From formula (3) $H_h = QD_2C_p(t_3 - t_2)$ and

$$Q = \frac{H_h}{D_2C_p(t_3 - t_2)} = \frac{860{,}000}{0.07524 \times 0.2415(140-68)}$$

$$= 657{,}000 \text{ cubic feet per hour.}$$

Since the amount of air required for ventilation was set at 600,000 cubic feet per hour, it is evident that the amount introduced for heating requirements will be ample for ventilation.

Now, assume that instead of 400 people, there are 500 to be provided for, requiring 750,000 cubic feet per hour. The 657,000 cubic feet demanded by the heating requirements will then be insufficient and the quantity delivered must be that required for ventilation, its temperature, t_3, being below 140°. The temperature, t_3, may be computed from equation (3).

$$860{,}000 = 750{,}000 \times 0.07524 \times 0.2415(t_3 - 68)$$

$$t_3 = 131°$$

190. Flow of Air in Ducts.—When air, like other fluids, is moved through a pipe or duct, a certain pressure or head is necessary to start and maintain the flow. This head has two components. The **static head** is that which is required to overcome the frictional resistance of the air against the surface of the duct. The **velocity head** is the pressure required to produce the velocity of flow. The sum of these two components is termed the **total** or **dynamic** head.

The static and velocity heads are mutually convertible. The velocity head depends entirely upon the velocity of flow and if the velocity in the duct is decreased at any point because of an increase in the cross-sectional area, a portion of the velocity head

will be converted into static head. Conversely, when the area is reduced, the static head is partially converted into velocity head. The interchange, however, is always accompanied by a certain amount of net loss of head, depending upon the abruptness of the change in area and shape of the section in which the change of area takes place.

The velocity head may be considered as the height of a column of air which will have at its base a pressure sufficient to produce the given velocity, the relation being represented by the common expression, $v^2 = 2gh$. To express the velocity head in inches of water, the usual standard, let

D = density of air under the given conditions, pounds per cubic foot.
D' = density of water = 62.3 pounds per cubic foot at 70°.
h_v = velocity head in inches of water.
h = velocity head in feet of air.

Then

$$hD = \frac{h_v}{12} D' \qquad \text{or} \qquad h = \frac{h_v}{12} \frac{D'}{D}$$

$$V^2 = 3600 \times 2g \times \frac{h_v D'}{12D}$$

in which V is the velocity in feet per minute.

$$V = 1096.5 \sqrt{\frac{h_v}{D}} \tag{1}$$

$$h_v = \left(\frac{V}{1096.5}\right)^2 D \tag{2}$$

191. Measurement of Flow.—The static head or pressure in an air duct may be thought of as the pressure tending to burst the duct and it may therefore be readily measured by means of a water gage communicating with the duct in the manner shown at A in Fig. 135. The deflection of the water levels will then indicate the static pressure directly in inches of water. The total or dynamic head is measured by a tube whose open end points against the flow as at B. Since the velocity varies at different points in the cross-section of the duct, any single reading of the total pressure applies only to the particular location of the tube in the duct. The velocity head, which is equal to the difference between the total and static heads, can

be computed from them or can be measured directly by connecting the U-tube as at C in Fig. 135.

The relation between the velocity and the velocity head affords a convenient method for measuring the flow of air through

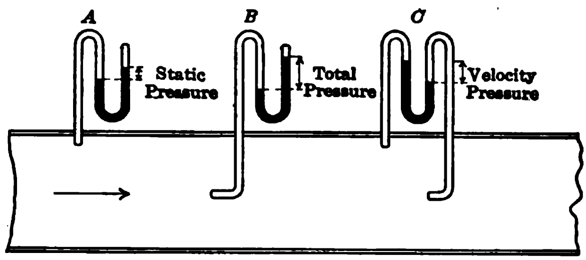

FIG. 135.

pipes. For this purpose the pitot tube illustrated in Fig. 136 is used in practice. The tube is inserted into the pipe in such a manner that the head A–B is parallel to the flow of air, with the end A toward the flow. The part A–B consists of an inner

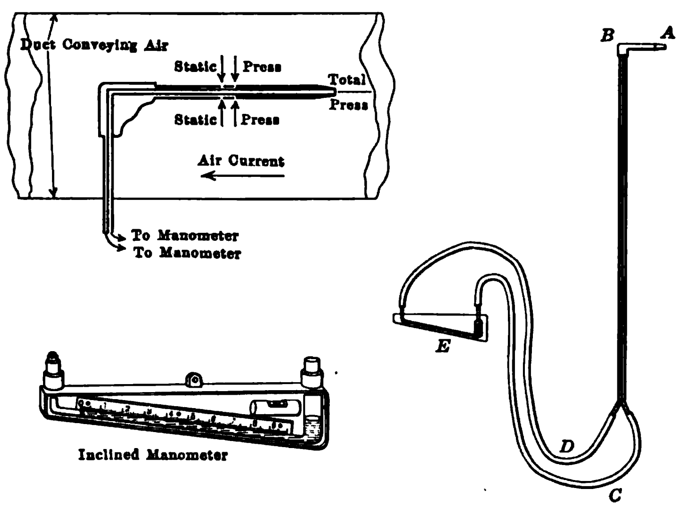

FIG. 136.—Pitot tube.

tube which transmits the total pressure to the tube D and an outer jacket through which the static pressure is transmitted to the tube C. This outer jacket contains several small holes through which the static pressure is transmitted. The two

pressures are transmitted to the ends of the differential slant gage *E*, which is a U-tube arranged with one leg at an angle so that the linear deflection per inch of height is increased. Gages of this type are usually filled with oil but are calibrated to read in inches of water column. The reading on the gage is evidently the velocity head, being the difference between the static and total heads.

As has been stated, the velocity of flow is not constant at all points in the cross-section of the duct. Near the walls it is retarded by the friction and it reaches a maximum at the center of the pipe. It is therefore necessary to measure the velocity at several points in the pipe in order to obtain an average figure. In a square or rectangular duct the cross-section is divided into

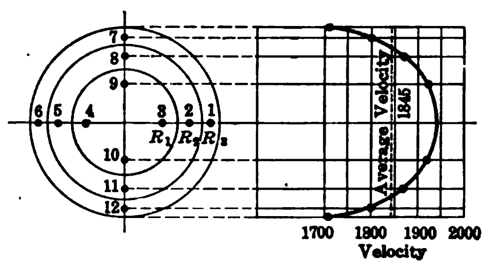

FIG. 137.—Division of round pipe into annular zones.

several equal rectangles and readings are taken with the pitot tube at the center of each of these divisions. The velocity corresponding to the pressure at the point where each reading is taken is then computed from formula (1), Par. 190, in feet per minute. The average of these computed velocities is taken as the average velocity in the pipe. The quantity of air flowing can be readily computed from the velocity and the cross-sectional area of the pipe.

For a round pipe the cross-sectional area should be divided into a number of annular zones of *equal area* and a traverse of the pipe should be made in both a vertical and a horizontal direction, as shown in Fig. 137. For each foot of pipe diameter the cross-section should be divided into at least three of these zones. Table XLI gives the distance from the center of the pipe at which each reading should be taken in per cent. of the pipe diameter. It is important that the velocities be computed separately and averaged, for the velocity varies as the square root of the pressure

TABLE XLI.—PIPE TRAVERSE FOR PITOT TUBE READINGS

Distance from Center of Pipe to Point of Reading in Per Cent. of Pipe Diameter

No. of equal areas in traverse	No. of readings	1st R_1	2d R_2	3d R_3	4th R_4	5th R_5	6th R_6	7th R_7	8th R_8
3	12	20.4	35.3	45.5					
4	16	17.7	30.5	39.4	46.6				
5	20	15.5	27.2	35.3	41.7	47.4			
6	24	14.5	25.0	32.3	38.2	43.3	47.9		
7	28	13.4	23.1	29.9	35.3	40.1	44.3	48.2	
8	32	12.5	21.6	28.0	33.2	37.6	41.5	45.1	48.4

and accurate results cannot be obtained by averaging the pressure readings. The method outlined above is the standard method adopted by the American Society of Heating and Ventilating Engineers.[1]

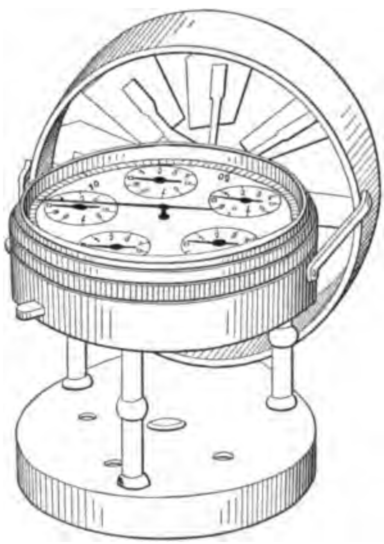

FIG. 138.—Anemometer.

192. The Anemometer.—For very approximate results, the anemometer, Fig. 138, is a convenient instrument for measuring the flow of air at the duct outlets. For very low velocities it is not suitable, as the friction required to revolve the propeller is

[1] Report of Committee on Standardization of Use of Pitot Tube, *Trans.* A. S. H. & V. E., 1914.

the source of a considerable error. In using the anemometer the face of the register is divided into a number of equal areas and the readings taken at the several areas are averaged. The dial is calibrated to read directly in feet and the velocity is obtained by taking the registration of the instrument during a definite period of time.

193. Friction Loss.—The general expression for the friction of fluids in pipes (equation (3), page 132) is applicable to the flow of air:

$$P = f\frac{S}{a}D\frac{v^2}{2g}$$

or for round ducts of perimeter R and length L

$$P = \frac{fRL}{a}\frac{Dv^2}{2g} \text{ or } h_a = \frac{fRL}{a}\frac{v^2}{2g}$$

in which P = pressure required to overcome friction, pounds per square foot.
a = cross-sectional area of duct, square feet.
D = density of air, pounds per cubic foot.
v = velocity, feet per second.
f = coefficient of friction.
h_a = height in feet of a column of air equivalent to P.

It is more convenient to express the friction head in terms of inches of water. If the density of air at 70° be taken as 0.075 and the density of water as 62.3 pounds per cubic foot then the head in inches of water is

$$h = \frac{0.075 \times 12}{62.3}\frac{fRL}{a}\frac{v^2}{2g} = 0.00022\frac{fRL}{a}v^2$$

The value of f was found by Reitschel and other to be about 0.0037 for smooth iron ducts. Prof. J. E. Emswiler[1] reports values for f ranging between 0.004 and 0.006 for velocities of 800 feet per minute and over, the coefficient decreasing slightly as the velocity increases. For practical purposes a somewhat higher coefficient is used, giving larger duct sizes. Allowance is thereby made for roughness of the duct surfaces and for accidental obstructions.

The chart in Fig. 139, which is published by the American

[1] See "Coefficient of Friction of Air Flowing in Round Galvanized Iron Ducts," by J. E. Emswiler, *Trans.* A. S. H. & V. E., 1916.

Blower Co., gives the friction in inches of water per 100 feet length of duct for various quantities of air. The chart is for round ducts; to figure the friction in a square or rectangular duct, it is necessary first to obtain the diameter of the equivalent round duct, which can be done by means of Table XLII.

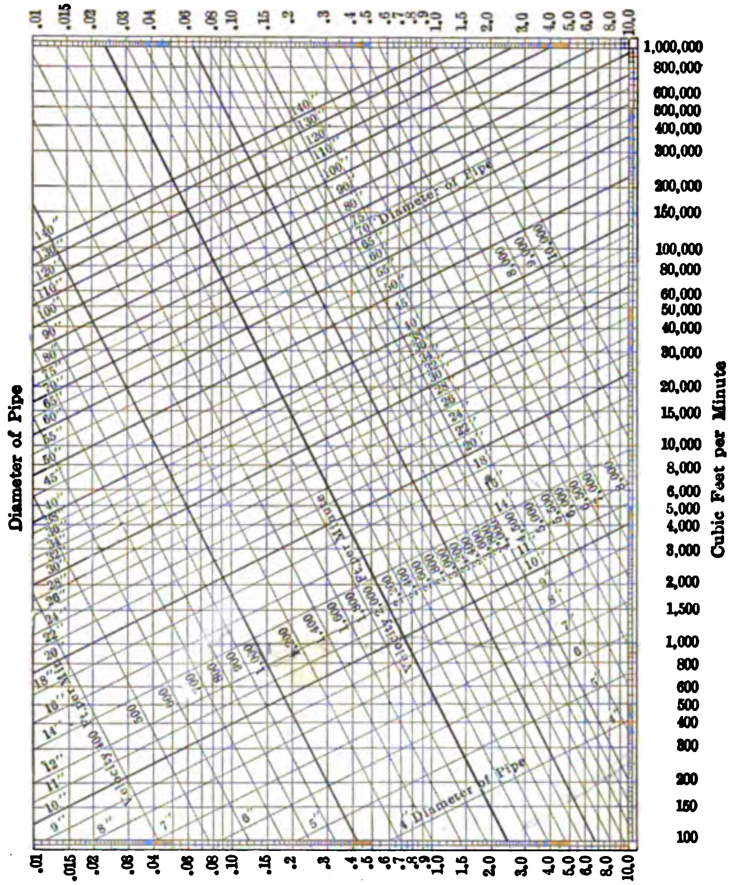

FIG. 139.—Frictional resistance in round air ducts.

Example.—Find the friction loss in a 20 by 10-inch duct 67 feet long, carrying 2000 cubic feet of air per minute. From Table XLII we find that the diameter of the equivalent round duct is 15.4 inches. From the chart in Fig. 139 the friction drop per 100 feet of duct for the given flow and for a diameter of 15.4 inches is readily found to be 0.31 inches of water. For a length of 67 feet the drop would be $0.3 \times 0.67 = 0.201$ inches of water.

TABLE XLII.—DIAMETER OF ROUND DUCTS EQUIVALENT TO RECTANGULAR DUCTS OF VARIOUS DIMENSIONS

Side rectangular duct	4	6	8	10	12	14	15	16	18	20	22	24
	Equivalent diameters											
3												
4	4.4											
5	4.9											
6	5.4	6.6										
7	5.8	7.0										
8	6.1	7.6	8.8									
9	6.5	8.0	9.3									
10	6.8	8.4	9.8	11.0								
11	7.1	8.8	10.2	11.5								
12	7.4	9.2	10.7	12.0	13.2							
13	7.6	9.6	11.1	12.5	13.7							
14	7.6	9.9	11.5	12.9	14.3	15.4						
15	8.2	10.2	11.9	13.4	14.7	16.0	16.5					
16	8.4	10.5	12.3	13.8	15.2	16.5	17.1	17.6				
17	8.6	10.8	12.6	14.2	15.7	17.0	17.6	18.2				
18	8.9	11.1	13.0	14.6	16.1	17.4	18.1	18.7	19.8			
19	9.1	11.4	13.3	15.0	16.5	17.9	18.6	19.2	20.4			
20	9.3	11.6	13.6	15.4	17.0	18.4	19.0	19.7	20.9	22.0		
22	9.7	12.1	14.2	16.1	17.8	19.2	19.9	20.6	21.9	23.1	24.2	
24	10.0	12.6	14.8	16.8	18.5	20.0	20.8	21.5	22.8	24.0	25.2	26.4
26	10.4	13.1	15.4	17.3	19.2	20.8	21.6	22.3	23.8	25.1	26.3	27.5
28	10.8	13.5	15.9	18.0	19.8	21.5	22.4	23.1	24.6	26.0	27.3	28.5
30	11.0	13.9	16.4	18.5	20.5	22.2	23.1	23.9	25.4	26.8	28.2	29.5
32	11.3	14.3	16.9	19.1	21.1	22.9	23.8	24.6	26.2	27.7	29.1	30.5
34	11.6	14.7	17.3	19.6	21.6	23.5	24.4	26.3	26.9	28.5	30.0	31.3
36	11.9	15.1	17.7	20.1	22.2	24.2	25.1	26.0	27.7	29.3	30.8	32.2
38	12.2	15.4	18.2	20.6	22.8	24.8	25.8	26.7	28.4	30.0	31.5	33.1
40	12.5	15.7	18.6	21.1	23.3	25.4	26.4	27.3	29.1	30.8	32.4	33.9
42	12.7	16.1	19.0	21.6	23.8	25.9	26.9	27.9	29.8	31.4	33.0	34.5
44	13.0	16.4	19.4	22.0	24.3	26.5	27.5	28.5	30.3	32.1	33.7	35.3
46	13.3	16.7	19.8	22.4	24.8	27.0	28.1	29.1	31.0	32.8	34.6	36.2
48	13.5	17.0	20.1	22.8	25.2	27.5	28.6	29.6	31.6	33.4	35.2	37.0
50	13.7	17.3	20.4	23.2	25.7	28.0	29.2	30.3	32.2	34.1	35.9	37.6
52	13.9	17.6	20.8	23.6	26.2	28.5	29.6	30.7	32.9	34.7	36.5	38.3
54	14.1	17.9	21.1	24.0	26.6	29.0	30.1	31.2	33.4	35.3	37.2	38.9
56	14.3	18.2	21.5	24.4	27.0	29.5	30.6	31.7	33.9	35.9	37.8	39.6
58	14.6	18.4	21.8	24.7	27.4	30.0	31.1	32.2	34.4	36.4	38.4	40.3
60	14.7	18.7	22.1	25.1	27.8	30.5	31.6	32.7	34.9	37.1	39.1	40.9
62	15.0	19.0	22.4	25.5	28.2	30.9	32.1	33.2	35.4	37.7	39.6	41.6
64	15.1	19.2	22.7	25.9	28.6	31.3	32.6	33.7	35.9	38.2	40.2	42.2
66	15.3	19.5	23.0	26.2	29.0	31.7	33.0	34.2	36.4	38.7	40.8	42.8
68	15.5	19.7	23.3	26.5	29.4	32.1	33.4	34.7	36.9	39.2	41.4	43.4

194. Pressure Loss Due to Obstructions.—The loss of pressure caused by various obstructions, such as elbows, branches, etc., is usually expressed as a multiple of the velocity head. The actual loss, however, is of course a loss of static head, since the velocity head at all points in a pipe, for a given quantity of

air flowing, depends entirely upon the cross-sectional area at each point.

The center line radius of elbows should be equal to at least 1½ times the width of the duct, as demonstrated by Frank L. Busey,[1] who obtained the following results for elbows of square cross-section:

Center line radius in per cent. of pipe width	Per cent. of velocity head lost
50	95
75	34
100	17
150	8
200	7

Another method is to add to the actual length of straight pipe a certain length which will have the same friction loss as that due to the resistance in question. The following table gives the loss of pressure due to various obstructions.

TABLE XLIII.—PRESSURE LOSS DUE TO VARIOUS OBSTRUCTIONS

	Per cent. of velocity pressure	Equivalent length of straight pipe
Round elbow (c. l. radius 1½ × width)	8–10	10 × width
Sharp elbow	100.0	
Square tee	100.0	
Branch from main duct		
Angle, 15 degrees (per cent. of v. p. in branch)	10	
30 degrees	20	
45 degrees	25	
Abrupt entrance to pipe	50–90	
Coned entrance to pipe	25	
Registers (free area = duct area = ½ total area of register).	1.25	

Air washers:

Velocity through washer, feet per minute	Pressure loss, inches of water
400	0.15
500	0.25
600	0.35
700	0.45

Example.—Given an air duct of square cross-section carrying air at a velocity of 900 feet per minute, and at a temperature of 70°. Find the loss

[1] See "Loss of Pressure Due to Elbows in the Transmission of Air through Pipes or Ducts," by FRANK L. BUSEY, *Trans.* A. S. H. & V. E., 1913.

of head due to an elbow of diameter $1\frac{1}{2} \times$ width. From formula (2), page 210, the velocity head $= \left(\frac{900}{1096.5}\right)^2 \times 0.07495 = 0.0505$ inches. The pressure loss is $0.08 \times 0.0505 = 0.004$ inches.

195. Proportioning Duct Systems.—It is highly desirable that the size of the ducts be intelligently selected and that the pressure loss in the system be computed as accurately as possible. The principal reason for doing this is to insure the selection of a fan of the proper characteristics; for in order that the required quantity of air be delivered it is necessary that a fan be selected with working head sufficient to overcome the resistance of the system. Furthermore, the proper proportioning of the various branches will result in the delivery of the proper air quantities to the various rooms without too great a dependence upon the use of the dampers.

In designing a duct system it is necessary first to select the static resistance against which the fan is to operate. Since the power consumption depends upon the resistance, the cost of power is a consideration in air-duct design. A reduction in the power required can be obtained by increasing the duct sizes; but the increased cost of the larger ducts and the greater space required are the opposing factors.

There are two general systems of air distribution and the method of choosing the duct sizes depends upon the type of system. In public buildings, particularly in schools, the single-duct system is used, in which the air is delivered to a plenum chamber by the fan and separate ducts radiate to the various rooms. In such a system the duct having the greatest resistance is first designed, which fixes the pressure to be carried in the plenum chamber. The other ducts are then so designed as to deliver the required quantities with the given pressure differential.

The longest duct is designed on a basis of certain assumed velocities; Table XLIV gives those recommended by W. H. Carrier:

TABLE XLIV.—VELOCITIES IN SINGLE-DUCT SYSTEMS

	Velocity, feet per minute
Vertical flues	400–750
Horizontal runs	700–1200
Wall registers[1]	200–400
Floor registers[1]	125–175

[1] Over gross area.

In industrial buildings the trunk duct system is used, consisting of one or more main ducts with branches taken off at intervals. Such ducts are so proportioned as to give an equal friction loss per foot of length. The outlets are designed for certain velocities depending upon the size of the room and upon the distance through which it is desired to blow the air, the possibility of objectionable drafts being considered. It is customary to assume an outlet velocity of from 700 to 1500 feet per minute, an average figure being 1000 feet per minute. The branches from the main duct should be so proportioned as to deliver the required air quantities and it is usually best to provide dampers on the outlets so that any inequalities in distribution can be adjusted after the system is installed. It is desirable to design all air ducts on a basis of an air density corresponding to the maximum air temperature to be expected.

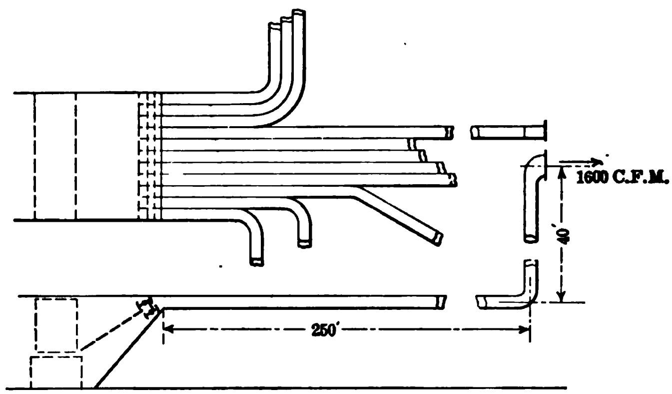

FIG. 140.

196. Example of Single Duct System.—Assume that a single duct system is to be designed and that the longest duct is arranged as in Fig. 140, the air temperature being 70°.

We will figure the horizontal run on a basis of 1000 feet per minute and a duct of rectangular section will be used. The area of the horizontal duct will be $1600 \div 1000 = 1.6$ square feet and a 12- by 19-inch duct will be used. For the riser a velocity of 600 feet per minute will be used and the required area is $1600 \div 600 = 2.75$ square feet, requiring a 16- by 24-inch duct. From Table XLII we find that the diameter of a round pipe equivalent to a 12- by 19-inch rectangular duct is 16.5 inches and for a 16- by 24-inch duct 21.5 inches. From the chart in Fig. 139 we find

that a pipe of 16.5 inches diameter will transmit 1600 c.f.m. with a friction loss of 0.14 inches per 100 feet, and the loss for a 21.5-inch pipe is 0.034 inches per 100 feet. To the actual length of straight pipe we must add the equivalent of the elbows, which may be taken (see Table XLIII) as ten times the actual width of the duct measured on the radius of the elbow. The total friction drop due to the straight pipe is then as follows:

$$(250 + 10) \times \frac{0.14}{100} + (40 + 13.3) \times \frac{0.034}{100} = 0.382 \text{ inch}$$

The resistance of the register may be taken as 1.25 times the velocity head corresponding to a register velocity of 300 feet per minute, upon which basis the size of the register will be selected. The velocity head we may compute by means of formula (2), page 210,

$$h_v = \left(\frac{300}{1096.5}\right)^2 \times 0.07495 = 0.0056 \text{ inch}$$

The loss through the register is $0.0056 \times 1.25 = 0.007$ inch. The loss at the entrance to the duct from the plenum chamber we will take as 80 per cent. of the velocity head corresponding to the velocity of 1000 feet per minute.

$$0.80 \times h_v = 0.80 \times \left(\frac{1000}{1096.5}\right)^2 \times 0.07495 = 0.050 \text{ inch.}$$

The total resistance of the duct is then

$$0.382 + 0.007 + 0.050 = 0.439 \text{ inch}$$

and the total pressure in the plenum chamber must be equal to this plus the velocity head corresponding to 1000 feet per minute or $0.439 + 0.062 = 0.501$ inch. The remaining ducts must then be of such a size as to use up this available total pressure of 0.501 inch.

Assume the following data for one of the ducts:

Quantity of air delivered,	1150 c.f.m.
Register velocity,	300 feet per minute.
Velocity, throughout entire length,	800 feet per minute.
Total equivalent length, including resistance of elbows,	110 feet.

The following quantities can be computed:

$$\text{Resistance of register} = \left(\frac{300}{1096.5}\right)^2 \times 0.07495 = 0.0056 \text{ inch.}$$

$$\text{Loss at entrance to duct} = 0.80 \times \left(\frac{800}{1096.5}\right)^2 \times 0.07495 = 0.032 \text{ inch.}$$

$$\text{Velocity head at entrance} = \left(\frac{800}{1096.5}\right)^2 \times 0.07495 = 0.040 \text{ inch.}$$

Static head to be used up by friction = 0.501 − (0.0056 + 0.032 + 0.040) = 0.423 inch.

The friction loss per 100 feet of duct must then be 0.423 ÷ 1.10 = 0.385 inch. From the chart in Fig. 139 the diameter of the round pipe which will give this friction loss for 1150 c.f.m. is 12.0 inches. This is equivalent (see Table XLII) to a rectangular pipe 10 by 12 inches or 8 by 15 inches, either of which could be used. The equivalent length allowed for the elbows, which must necessarily have been estimated, should be revised if the computed width of the duct is greatly different from the assumed width upon which the equivalent lengths were estimated, and the calculation repeated.

197. Correction for Temperature.—The quantities for which the duct sizes are computed are the volumes at the actual temperature of the air flowing. On the other hand, the volumes fixed by the heating and ventilating requirements are on a basis of room temperature, *i.e.*, about 70°. The volumes upon which the air ducts are designed must therefore be determined by multiplying the volumes at 70° by the ratio:

$$\frac{\text{Density of air at } 70^\circ}{\text{Density of air at duct temperature}}$$

These ratios are given in Table XXXVI, page 176, in the column headed "Ratio to Volume at 70°F."

198. Trunk-line System.—In a trunk-line system, the procedure would be as follows:

Assume a system laid out as in Fig. 141, in which the quantities as given are on a basis of 70°. The system will be designed for a temperature of 135° and the actual quantities flowing in the various sections are as follows;

A–B	11,100 × 1.1230	= 12,465
B–C	5,800 × 1.1230	= 6,513
C–D	1,800 × 1.1230	= 2,021
B–E	3,300 × 1.1230	= 3,706
E–F	1,500 × 1.1230	= 1,684

The total head at point *A* must be equal to the friction loss in the trunk duct plus the velocity head at *D*, the end of

the trunk duct. The method of proportioning by a uniform friction loss leads to a reduction in the velocity toward the end of the trunk and a consequent conversion of some of the velocity head to static head. The absolute values of the velocity and static heads at A are not important, the requirement being that their sum be equal to the friction loss plus the velocity head at D. On a basis of velocity of 1000 feet per minute the velocity head at D will be equal to $\left(\frac{1000}{1096.5}\right)^2 \times 0.06675 = 0.055$ inch on a basis of 135°. The friction drop may be fixed arbitrarily and we will choose it in this case as 0.20 inch per 100 feet, giving a total pressure at point A of $0.20 \times 2.25 + 0.055 = 0.505$ inch. For a friction drop of 0.20 inch per 100

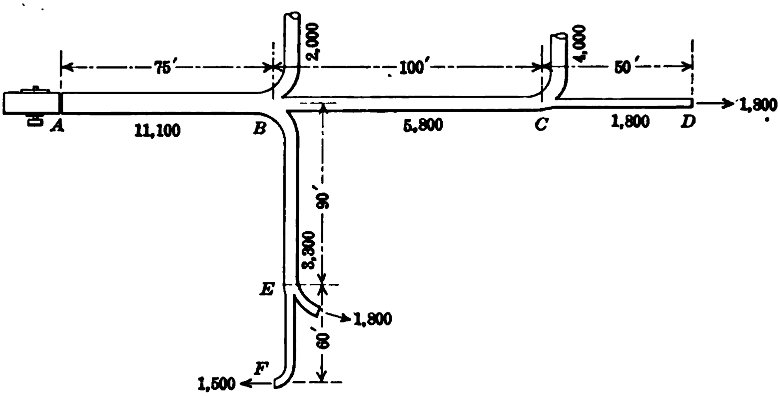

FIG. 141.

feet the diameters of sections A–B, B–C, and C–D, would be respectively 34.0, 26.0, and 17 inches. The diameter of the outlet at D would be increased to 19 inches to give the required outlet velocity of 1000 feet per minute. The branch pipe could be designed for the same pressure loss per unit length but it is more economical to take advantage of the full available head and reduce the size of the pipe. The static head at B can be found by subtracting from the static head at A the loss in section A-B. Allowing for the loss due to entrance in the branch at B and for the final velocity head at F the allowable friction loss in sections BE and EF can be determined and the size of pipe chosen accordingly. All outlets should be provided with dampers so that the proper delivery can be obtained by adjusting them after the system is installed.

199. Power Required for Moving Air.—The power required for moving air through a system of ducts may be expressed as follows:

Let p = unit total pressure, inches of water.
a = cross-sectional area of duct, square feet.
v = velocity of air, feet per minute.

Then the horsepower required is

$$Hp. = \frac{pav \times 144}{12 \times 2.31 \times 33{,}000} = 0.000158\,pav \qquad (1)$$

If q is the volume of air delivered per minute in cubic feet, then $q = av$ and

$$Hp. = 0.000158\,pq$$

200. Theory of the Centrifugal Fan.—The centrifugal fan consists fundamentally of a wheel having several radial vanes revolving in a casing. Air enters near the axis of the wheel, flows to the circumference under the influence of the centrifugal force produced by the rotation, and is discharged through the outlet which is located tangentially with respect to the fan wheel. The pressure created in a fan has two separate and independent sources, (*a*) that due to the centrifugal force imparted to the masses of air enclosed between the vanes, and (*b*) the pressure due to the linear velocity of the air as it leaves the tip of the blades. The efficient conversion of the velocity head into static head depends upon the proper design of the fan housing, as will be shown later.

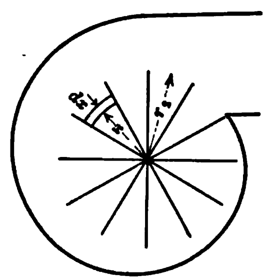

FIG. 142.

Fig. 142 represents an elementary centrifugal fan. Consider a thin layer of air of thickness dx between two of the vanes at a distance x from the axis and having an area of S. The volume of this layer of air is then Sdx, and if its density is D, then the weight will be $SdxD$. Assume that the fan outlet is completely closed and that the wheel revolves at the rate of ω radians per second. Then the centrifugal force

$$df = \frac{\omega^2 xSdxD}{g}\text{[1]}$$

[1] Centrifugal force $= \frac{m\omega^2 r}{g}$ for a mass m at radius r.

The unit pressure dp corresponding to df is evidently $= \frac{df}{S}$ and the equivalent head

$$dh = \frac{dp}{D} = \frac{df}{SD}.$$

Then
$$dh = \frac{\omega^2 x dx}{g}$$

Let r_1 be the radius at the base of the blade and r_2 the radius at the tip. Then

$$h = \int_{r_1}^{r_2} \frac{\omega^2 x dx}{g} = \frac{\omega^2 r_2{}^2 - \omega^2 r_1{}^2}{2g}$$

If the entire column of air between the two blades from the axis to the radius r_2 be assumed to be affected, then $r_1 = 0$ and

$$h = \frac{\omega^2 r_2{}^2}{2g}$$

If v is the linear tip speed then $v = \omega r_2$ and

$$h = \frac{v^2}{2g}$$

The second source of pressure is that equivalent to the velocity v of the air at the blade tips which is equal to

$$h' = \frac{v^2}{2g}$$

The total pressure or head developed under the assumed conditions would then be

$$h + h' = \frac{v^2}{g}$$

The above analysis is approximate only and is complicated under actual conditions by the effect of the various sources of pressure loss and by the fact that the conversion of the velocity head into static head is only partial. The analysis serves to show, however, the relation between the pressure developed by a centrifugal fan and the fan speed.[1]

201. Fan Blades and Housings.—Fan blades may be designed in either of three ways: radial, curved, forward (*i.e.*, in the direction of rotation) or curved backward. In Fig. 143 is shown

[1] For a complete discussion of the subject see "Heating and Ventilating of Buildings," by R. C. CARPENTER.

graphically the effect in the resultant velocity of the air due to the different blade designs. The air leaving the tip of the blade has a velocity component v_1, parallel to the blade and a tangential component v_2. If the blade is curved forward the resultant velocity v will be greater than that in the straight-blade type and if curved backward the resultant velocity will be decreased.

The velocity head developed by the fan wheel is considerably greater than is required, while the static head, which is the force necessary to move the air against the frictional resistance of the duct system is low. The velocity head is therefore partially converted into static head by designing the housing in a suitable scroll shape so that the velocity of the air is gradually reduced. The efficiency with which the conversion to static head takes place depends upon the proper design of the housing. It is the

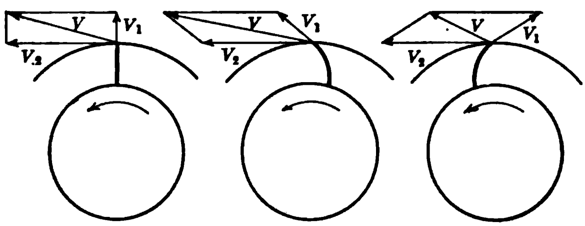

FIG. 143.—Effect of various blade designs.

static head developed by a fan which is useful in overcoming duct resistance and before the velocity head can become available it must be converted into static head. Generally speaking, the fan which has the greater static head in proportion to velocity head is the more desirable; although the velocity head may be further converted to static head after it leaves the fan if the velocity is reduced by a gradual enlargement of the duct area.

202. Power Required by a Fan.—It has been shown that the power required for moving air is

$$Hp. = \frac{pq \times 144}{12 \times 2.31 \times 33{,}000}$$

in which the pressure p is expressed in inches of water.
If the pressure is expressed in terms of the equivalent column of air of height h, then

$$Hp. = \frac{hDQ}{33{,}000}$$

in which D is the density of the air in pounds per cubic foot.

In a fan the actual head developed is only a portion of the theoretical head $\frac{v^2}{g}$ and is represented approximately by $\frac{kv^2}{g}$. The power required to drive a fan is then

$$Hp. = \frac{ckv^2}{g} \times \frac{DQ}{33{,}000}$$

in which c is a factor which takes into account the mechanical losses in the fan. Combining all of the constant factors we have

$$Hp. = Kv^2QD$$

v being the peripheral velocity, which varies directly as the speed of the fan. Since Q varies directly as the speed, the power required varies as the cube of the speed.

203. Fan Performance.—From a consideration of the foregoing, the following laws can be stated as to the performance of centrifugal fans:

For a given fan delivering air through a given piping system—

1. *The capacity varies directly as the fan speed.*
2. *The pressure varies as the square of the speed.*
3. *The speed and capacity vary as the square root of the pressure.*
4. *Horsepower varies as the cube of the speed or capacity.*
5. *Horsepower varies as the* $(pressure)^{3/2}$

For a constant pressure—

6. *The speed, horsepower and capacity vary as the square root of the absolute temperature of the air.*

At constant capacity and speed—

7. *The horsepower and pressure vary inversely as the absolute temperature of the air.*

204. Fan Efficiency.—The true efficiency of a fan may be defined as the ratio of the work done in moving the air to the energy input to the fan. The **total efficiency** which is the true efficiency is computed from the total pressure, while the so-called **static efficiency** is computed from the static pressure. The efficiency may then be expressed as follows:

$$\text{Static efficiency} = \frac{0.000157 \times \text{c.f.m.} \times \text{static pressure in inches}}{\text{hp.}}$$

$$\text{Total efficiency} = \frac{0.000157 \times \text{c.f.m.} \times \text{total pressure in inches}}{\text{hp.}}$$

in which $hp.$ represents the horsepower input to the fan, and the

factor 0.000157 is the power required to move 1 cubic foot of air per minute against a pressure of 1 inch of water.

205. Straight-blade and Multi-blade Fans.—Centrifugal fans are of two general types. The older type, the "steel-plate" fan, has a relatively small number of radial blades which are nearly plane surfaces. The more recently developed "multi-blade" type has a large number of short, curved blades on a wheel of comparatively small diameter. In the multi-blade type the blades are usually curved forward as in Fig. 143, so that the pressure

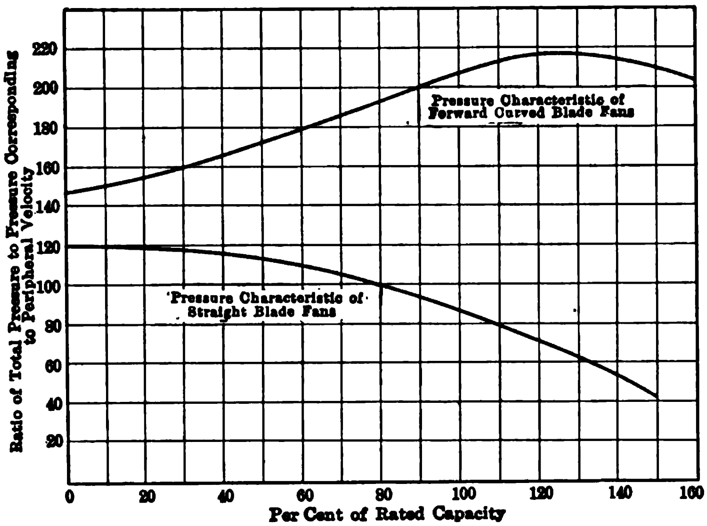

FIG. 144.—Pressure characteristics of straight-blade and multi-blade fans at constant speed.[1]

developed will be greater than that corresponding to the peripheral velocity.

The two types of fans have inherently different characteristics. In a straight-blade fan operated at constant speed the total pressure developed decreases as the output of the fan is allowed to increase by reason of a lessened resistance. The multi-blade fan develops an increasing total pressure as its output is increased under the same conditions. In Fig. 144 are shown the pressure characteristics of the two types. The vertical ordinate is in terms of the ratio of the total pressure to the pressure corre-

[1] From "The Centrifugal Fan," by FRANK L. BUSEY, *Trans.* A. S. H. & V. E., 1915.

sponding to the peripheral velocity, this standard being used simply to make the curves comparable. The practical significance of these differing characteristics is evident when the action of a fan supplying a system of ducts is considered. With a straight-blade fan if one part of the duct system were shut off and the fan speed is unchanged the result would be an increase in the amount of air delivered to the other rooms. With a multi-blade fan, on the other hand, the quantity delivered through the remaining ducts would not be greatly altered. Other advantages of fans of the multi-blade type are the smaller space occupied and the fact that their higher speed makes it possible to connect them direct to motors. The higher speed also reduces the cost of the motor in some cases. In general the multi-blade type is the more suitable for ventilating systems.

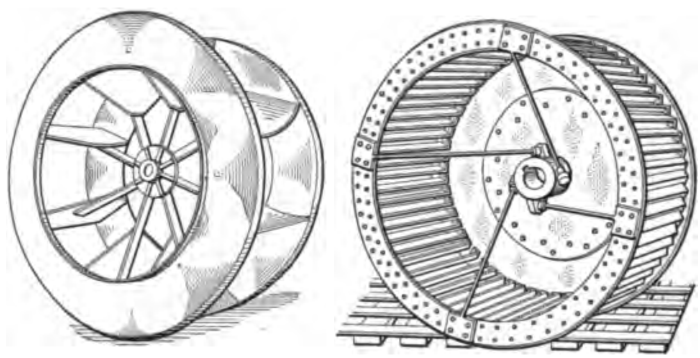

FIG. 145.—Wheel of straight-blade fan. Wheel of multi-blade fan.

206. Commercial Types.—In Fig. 145 are shown the wheels of a straight-blade and of a multi-blade fan and in Fig. 146 is shown the casing of a multi-blade fan. The general appearance of the casings of the two types is quite similar, the multi-blade fan being somewhat smaller in diameter and of greater width for the same capacity. Fans can be obtained with the discharge opening at various angles and with the inlet opening on either side. In some cases fans of double width, having an inlet on both sides, are used.

207. Selection of a Fan.—Before selecting a fan for a given installation it is necessary to know the quantity of air to be handled and the static resistance of the duct system. The total pressure against which the fan must operate is the sum of the

static resistances on both the suction and the discharge sides of the fan plus the velocity head at the fan outlet, which can be determined from the volume of air delivered and the size of the outlet. The size of fan which will fill the requirements is best obtained from the data published by the various fan manufacturers. It is usually possible to obtain the same capacity and static head from two or more different size fans. Frequently the fan which operates the most efficiently under the given conditions is not the lowest in first cost and the selection must be governed by the relative importance of these factors.

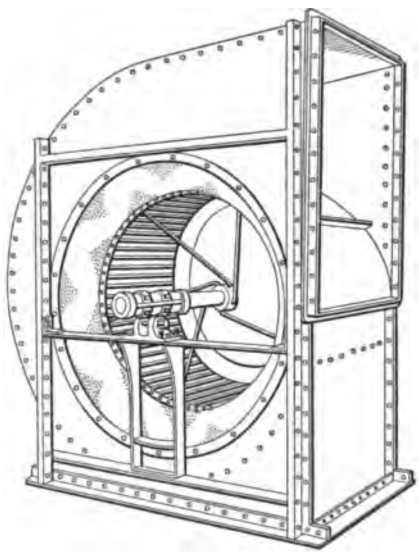

FIG. 146.—Casing of multi-blade fan.

208. Fan Tables.—The exact performance to be expected of a fan under any given conditions can be obtained from the tables published by the manufacturers. There are two kinds of fan tables—the "total pressure" tables, which give the speed, capacity, and horsepower for the various size fans *at the most efficient point* for various total pressures; and the more complete "static pressure" tables, which give the performance at points on either side of the most efficient point. Tables XLV and XLVI are, respectively, the total pressure table for various sizes of one type of multi-blade fan, and the static pressure table for a multi-blade fan of one particular size, the latter being in a some-

what condensed form. More complete static pressure tables for both steel plate and multi-blade fans may be found in the Ap-

Table XLV.—Capacities of Buffalo Niagara Conoidal Fans (Type N) Under Average Working Conditions—at 70°F. and 29.92 Inches Barom.[1]

Fan No.	Mean diam. of blast wheel	Area of outlet, square feet	⅜-in. total press. or 0.217 oz.			½-in. total press. or 0.288 oz.		
			R.p.m.	Vol.	Hp.	R.p.m.	Vol.	Hp.
3	15⅝	1.31	413	1,490	0.13	478	1,720	0.19
3½	18⅛	1.79	354	2,030	0.17	409	2,350	0.26
4	20½	2.33	310	2,650	0.22	358	3,070	0.34
4½	23½	2.95	276	3,360	0.28	318	3,880	0.43
5	26⅛	3.64	248	4,150	0.35	287	4,790	0.53
5½	28¾	4.41	225	5,020	0.42	260	5,800	0.65
6	31⅜	5.25	207	5,970	0.50	239	6,900	0.77
7	36½	7.14	177	8,130	0.68	205	9,400	1.05
8	42	9.33	155	10,610	0.89	179	12,260	1.37
9	47	11.81	138	13,450	1.12	159	15,520	1.73
10	52	14.58	124	16,580	1.39	143	19,160	2.14
11	58	17.64	113	20,070	1.68	130	23,180	2.58
12	63	21.00	104	23,880	2.00	119	27,590	3.08
13	68	24.65	95	28,040	2.35	110	32,370	3.61
14	73	28.68	89	32,520	2.72	102	37,550	4.19
15	78	32.80	83	37,330	3.13	96	43,100	4.80

Static pressure is 77½ per cent. of total press.

Table XLV.—(*Continued*)

Fan No.	Mean diam. of blast wheel	Area of outlet, square feet	⅝-in. total press. or 0.360 oz.			¾-in. total press. or 0.433 oz.		
			R.p.m.	Vol.	Hp.	R.p.m.	Vol.	Hp.
3	15⅝	1.31	533	1,930	0.27	585	2,110	0.35
3½	18⅛	1.79	457	2,620	0.37	501	2,870	0.48
4	20½	2.33	400	3,430	0.48	439	3,750	0.63
4½	23½	2.95	356	4,340	0.60	390	4,750	0.80
5	26⅛	3.64	320	5,350	0.74	351	5,870	0.98
5½	28¾	4.41	291	6,470	0.90	319	7,100	1.19
6	31⅜	5.25	267	7,710	1.07	292	8,450	1.41
7	36½	7.14	229	10,490	1.46	251	11,500	1.92
8	42	9.33	200	13,700	1.91	219	15,020	2.51
9	47	11.81	178	17,340	2.41	195	19,000	3.18
10	52	14.58	160	21,400	2.98	175	23,460	3.93
11	58	17.64	146	25,900	3.60	160	28,390	4.75
12	63	21.00	133	30,820	4.29	146	33,780	5.65
13	68	24.65	123	36,180	5.03	135	39,650	6.63
14	73	28.68	114	41,950	5.84	125	45,990	7.69
15	78	32.80	107	48,160	6.70	117	52,790	8.83

Static pressure is 77½ per cent. of total press.

[1] From "Fan Engineering," Buffalo Forge Co.

pendix, pages 276 to 299. The static pressure tables are the better adapted for general use. The total pressure can be found for any conditions by adding to the static pressure the velocity pressure as given in the third column in Table XLVI.

TABLE XLVI.—No. 10 NIAGARA CONOIDAL FAN (TYPE N)

Capacities and Static Pressures at 70°F. and 29.92 Inches Barom.[1]

Outlet velocity, ft.-min.	Capacity, cu. ft., air per min.	Add for total press.	½-in. s.p.		¾-in. s.p.		1-in. s.p.		1½-in. s.p.		2-in. s.p.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1,400	20,410	0.122	**164**	**2.92**	206	4.61	243	6.59	308	11.1		
1,500	21,870	0.141	163	3.13	204	4.78	240	6.83	305	11.5		
1,600	23,330	0.160	164	3.42	202	5.02	238	7.05	302	11.8	357	17.0
1,700	24,790	0.180	165	3.74	**201**	**5.30**	235	7.28	299	12.1	353	17.5
1,800	26,240	0.202	166	4.13	**200**	**5.61**	233	7.59	295	12.4	350	17.9
1,900	27,700	0.225	168	4.55	200	6.01	232	7.91	293	12.7	347	18.3
2,000	29,160	0.250	171	5.04	200	6.48	**231**	**8.32**	291	13.0	343	18.7
2,100	30,620	0.275	174	5.56	201	7.00	231	8.77	288	13.5	340	19.2
2,200	32,080	0.302	177	6.12	203	7.54	230	9.31	286	13.9	338	19.6
2,300	33,540	0.330	180	6.76	205	8.16	231	9.92	285	14.4	336	20.1
2,400	34,990	0.360	183	7.43	207	8.86	232	10.60	**284**	**15.0**	332	20.6
2,600	37,910	0.422	190	8.95	213	10.40	235	12.10	282	16.3	329	21.8
2,800	40,830	0.489	198	10.70	219	12.20	240	13.90	283	18.1	**327**	**23.2**
3,000	43,740	0.560	206	12.70	226	14.30	246	16.00	285	20.1	326	25.0
3,200	46,660	0.638	215	14.80	234	16.70	251	18.30	288	22.4	327	27.4

NOTE.—Bold-face figures indicate point of highest static efficiency.

The fan tables are based on actual tests made by operating the fan at constant speed against different artificial resistances consisting of plates, having openings of various sizes, placed at the end of a straight pipe about 30 diameters in length. In Fig. 147 are shown the performance curves for a multi-blade fan, based on the percentage of rated capacity, the latter being taken as the point at which the fan operates with the highest total efficiency. It should be borne in mind that these performance curves are based on a *constant speed.*

It is frequently necessary to find the performance of a fan at some pressure different from any given in the tables. The method of doing this can best be shown by a typical example. Assume that 38,000 cubic feet of air per minute is to be delivered by a No. 10 Conoidal fan against a static resistance of 1¼ inches. Find the required speed and horsepower. The data for 1-inch static is given in Table XLVI. The corresponding capac-

[1] From "The Centrifugal Fan," by FRANK L. BUSEY, *Trans.* A. S. H. & V. E., 1915.

ity of the fan at 1-inch static may be found by multiplying by the square root of the ratio of 1-inch to 1¼-inch, since we know that the pressure varies as the square of the speed and consequently as the square of the volume delivered. The capacity on a 1-inch basis is thus found to be 34,100 c.f.m. From Table XLVI we find that the speed and horsepower for 33,540 c.f.m. at 1-inch static are respectively 231 r.p.m. and 9.92 horsepower.

The speed and horsepower at 1¼ inches static we can compute from our knowledge that the speed varies directly as the capacity and the power as the cube of the capacity. The fan will deliver

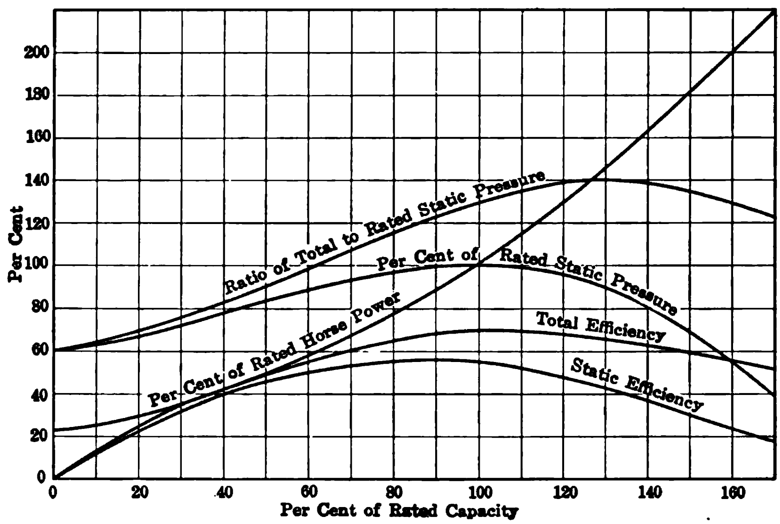

FIG. 147.—Performance curves of Niagara conoidal fans.

38,000 c.f.m. against 1¼ inches static with a speed of 258 r.p.m. and a power consumption of 13.9 horsepower.

In selecting a fan for a given installation it is usually possible to fulfill the required conditions with two or even three different-size fans. In such a case the first cost, operating cost, and outlet velocities should be considered in making the selection. The smaller the fan the greater will be the outlet velocity for the same volume. In the case of schools or other buildings where quiet operation is essential the outlet velocity should not be over about 2200 feet per minute. In industrial buildings, however, outlet velocities of about 3000 feet per minute are quite permissible.

209. Correction for Temperature.—The fan tables are based on an air density corresponding to a temperature of 70°. In a system in which the fan is so located with respect to the heating coils that it handles air at a different temperature, a correction must be made. This can be done by making use of the relations stated in Par. 203.

For example: Assume that it is required to handle 11,700 c.f.m. against a static head of 1¾ inches at 140°. As brought out in Par. 203, at constant capacity and speed, the horsepower and pressure vary inversely as the absolute temperature of the air. Therefore, if we select a fan which will handle 11,700 c.f.m. against a pressure of $1.75 \times \frac{600}{530} = 1.98$ inches at 70°, it will deliver the same quantity against a pressure of 1.75 inches at 140° at the same speed. From the fan tables we find that a No. 90 steel plate fan will do this at a speed of 403 r.p.m. and a power consumption of 7.32 horsepower. The power consumption at 140° would be $7.32 \times \frac{530}{600} = 6.46$ horsepower.

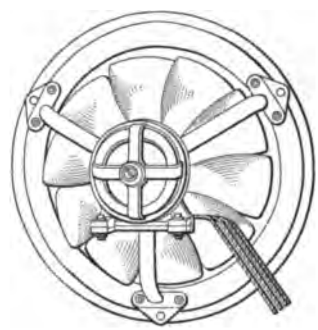

Fig. 148.—Disc fan.

It should be remembered that the volume of air fixed by the heating or ventilating requirements is usually based on the room temperature and the equivalent volume of the same weight of air at the temperature at which it enters the fan must be found by means of the volume ratios given in Table XXXVI, page 176.

210. Disc Fans.—The disc fan as illustrated in Fig. 148 is well adapted for handling considerable quantities of air against very low pressures. It is therefore widely used where the air is moved into or from a room without passing through a system of ducts. While not highly efficient, this type of fan is easily installed, is of moderate cost, and requires little space. Such a fan is usually inserted directly into a wall or partition and is driven by a direct-connected motor.

211. Heaters.—In a fan system the heat is transmitted from the heating units entirely by convection, the air being drawn over them at a fairly high velocity. There are two types of heater

used for such work—the cast-iron or "vento" heater and the wrought-iron pipe coil. The former is made up of sections, as shown in Fig. 149, connected together at the top and bottom by right- and left-hand nipples cast with a hexagonal nut at the middle. A row of sections thus connected constitutes a **stack.** The sections are obtainable in nominal lengths of 30, 40, 50, 60, and 72 inches. All sizes are connected at both top and bottom and are therefore suitable for hot water as well as steam. Vento sections are furnished in two widths, the "regular" and the "narrow," and by the use of nipples of different lengths the distance between sections can be made either $4\frac{5}{8}$, 5, or $5\frac{3}{8}$ inches center to center, the 5-inch spacing being standard. The surfaces are broken up by a large number of projections which extend into the air passages and serve to augment the heating surface in an effective manner. The principal dimensions of the sections of various sizes are given in Table XLVII.

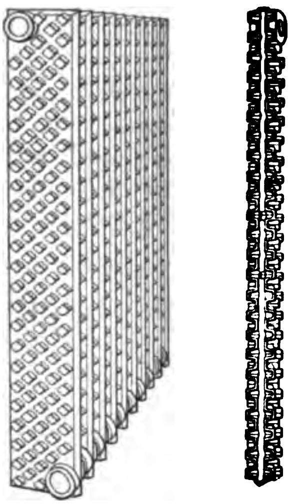

FIG. 149.—Vento heater.

TABLE XLVII.—DIMENSIONS OF VENTO SECTIONS, INCHES

Nominal size		Square feet of surface	Actual height	Width
Regular width	30	8.00	30	$9\frac{1}{8}$
	40	10.75	$41\frac{1}{64}$	$9\frac{1}{8}$
	50	13.50	$50\frac{9}{32}$	$9\frac{1}{8}$
	60	16.00	$60\frac{11}{16}$	$9\frac{1}{8}$
	72	19.00	$72\frac{3}{32}$	$9\frac{1}{8}$
Narrow	40	7.50	$41\frac{1}{64}$	$6\frac{3}{4}$
	50	9.50	$50\frac{29}{32}$	$6\frac{3}{4}$
	60	11.00	$60\frac{11}{16}$	$6\frac{3}{4}$

Approximate weight 8.2 pounds per square foot of surface.

The method of installing the stacks in a sheet-metal casing is shown in Fig. 150. The stacks are staggered so as to break up the stream lines and increase the intimacy of the contact between the air and the heating surface. The spaces left at the

ends of the stacks due to the staggered arrangement are partially closed by strips of angle iron.

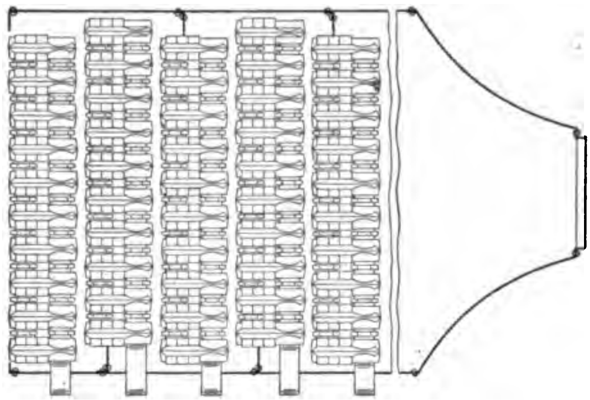

FIG. 150.—Vento heater installed in casing.

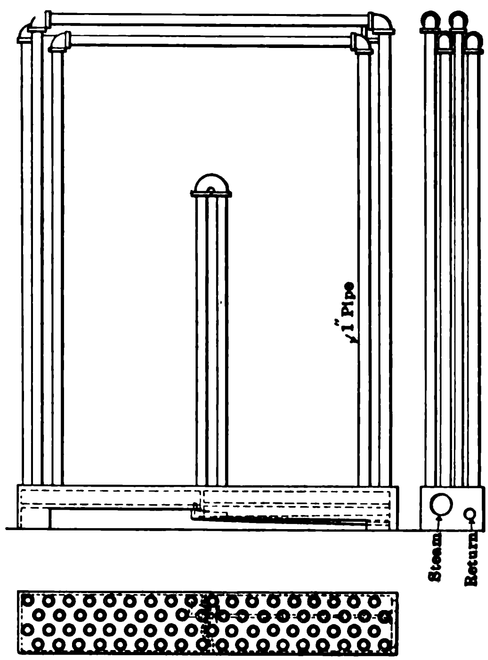

FIG. 151.—Pipe coil heater.

212. Pipe-coil Heaters.—Heaters made of 1-inch pipes are also widely used. The pipe is made into loops with ordinary

elbows, and the loops are screwed into a cast-iron base. The base is so partitioned that the steam flows in at one end of each of the loops. The sections are arranged as shown in Fig. 151, the pipes being staggered with reference to the flow of air through the heater. The sections are built in different sizes and a wide range in heating surface is available. The complete heater is composed of several units in series, as in the case of the cast-iron heaters.

213. Transmission of Heat From Fan-coil Surfaces.—The heating units are arranged in series, the outside air entering the first section and being heated up to the required delivery temperature during its passage through the successive sections. Since the rate of heat transmission varies nearly as the temperature difference between the steam and the air, the heat transmitted from the last stacks is much less than from those with which the cold air first comes into contact.

The final temperature to which the air is heated depends upon the number of stacks through which the air passes in series and upon the velocity of the air. The cross-sectional area of the heater depends upon the quantity of air delivered, the stacks being chosen of sufficient size so that the free area between the sections will allow that quantity to pass through at the velocity chosen. The free area per section for Vento heaters is given in Table XLVIII. Similar data is published by the manufacturers of pipe-coil heaters.

Table XLVIII.—Free Areas of Vento Sections

Size of section, inches	Free area, square inches per section		
	5⅝-in. centers	5-inch centers	4⅝-inch centers
30	0.542	0.460	0.390
40	0.729	0.620	0.525
50	0.905	0.768	0.650
60	1.085	0.921	0.781
72	1.303	1.104	0.937

The velocity to be assumed depends upon the nature of the installation. In public buildings and in other places where the noise which accompanies high velocities is objectionable, the velocity through the heater should be limited to between 1000 to 1300 feet per minute while in factories and similar buildings a

TABLE XLIX.—FINAL TEMPERATURES AND CONDENSATION

Regular Section—Standard Spacing, 5-inch Centers of Sections—Steam, 227°, 5 Pounds Gage

Number of stacks deep	Temperature of entering air	Velocity through heater in feet per minute—measured at 70°															
		600		800		1,000		1,200		1,400		1,600		1,800		2,000	
		Final temp. of air leaving heater	Cond. lb. per sq. ft. per hour	F.t.		F.t.	C.	F.t.	C.	F.t.	C.	F.t.		F.t.	C.	F.t.	C.
	−20																
	−10	34	1.69														
1	0	43	1.65	38	1.95	35	2.24	32	2.46								
	20	58	1.46	54	1.75	51	1.99	49	2.23	47	2.42	45	2.56	43	2.65	42	2.82
	30	66	1.39	62	1.64	60	1.92	58	2.17	56	2.33	54	2.46	52	2.54	51	2.69
	−20	63	1.60	55	1.92	49	2.22	44	2.46	40	2.69	37	2.92	34	3.12	31	3.27
	−10	69	1.52	62	1.85	56	2.12	51	2.35	47	2.56	44	2.77	41	2.94	38	3.08
2	0	75	1.44	68	1.74	62	1.99	58	2.23	54	2.42	51	2.62	48	2.77	46	2.95
	20	87	1.29	81	1.57	76	1.80	72	2.00	69	2.20	66	2.36	64	2.54	62	2.69
	30	93	1.21	87	1.46	83	1.70	79	1.89	76	2.06	73	2.21	71	2.37	69	2.50
	−20	91	1.42	82	1.74	75	2.03	69	2.28	64	2.51	59	2.70	55	2.88	52	3.08
	−10	96	1.36	87	1.66	80	1.92	75	2.18	70	2.39	66	2.60	62	2.77	58	2.90
3	0	101	1.30	93	1.59	86	1.84	81	2.08	76	2.27	72	2.46	68	2.62	65	2.78
	20	110	1.15	103	1.42	97	1.65	92	1.85	88	2.06	85	2.22	82	2.38	79	2.52
	30	115	1.09	108	1.33	103	1.56	98	1.75	94	1.91	91	2.08	88	2.23	85	2.35
	−20	114	1.29	103	1.58	96	1.86	90	2.12	84	2.34	78	2.51	74	2.71	70	2.88
	−10	117	1.22	108	1.51	101	1.78	95	2.02	89	2.22	84	2.41	80	2.60	76	2.76
4	0	121	1.16	113	1.45	106	1.70	100	1.92	95	2.13	90	2.31	86	2.48	82	2.63
	20	130	1.06	122	1.31	115	1.52	110	1.73	105	1.91	101	2.08	97	2.22	94	2.37
	30	134	1.00	126	1.23	120	1.44	115	1.63	110	1.80	106	1.95	102	2.08	99	2.21
	−20	132	1.17	122	1.46	114	1.72	107	1.95	100	2.15	94	2.34	90	2.54	86	2.72
	−10	135	1.13	126	1.40	118	1.64	111	1.86	105	2.06	99	2.24	95	2.42	91	2.59
5	0	138	1.06	129	1.32	122	1.56	115	1.77	109	1.96	104	2.14	100	2.31	96	2.46
	20	144	.95	136	1.19	130	1.41	124	1.60	119	1.78	114	1.93	110	2.08	107	2.23
	30	148	.91	140	1.13	134	1.33	128	1.51	123	1.67	118	1.80	115	1.96	112	2.10
	−20	146	1.06	137	1.34	129	1.59	121	1.81	115	2.02	110	2.22	105	2.40	100	2.56
	−10	149	1.02	140	1.28	132	1.52	125	1.73	119	1.93	114	2.12	109	2.29	104	2.44
6	0	152	.97	143	1.22	135	1.44	129	1.65	123	1.84	118	2.02	113	2.17	109	2.33
	20	156	.87	148	1.10	142	1.30	129	1.49	130	1.65	126	1.81	122	1.96	118	2.09
	30	159	.83	151	1.04	145	1.23	139	1.40	134	1.56	130	1.71	126	1.85	122	1.97
	−20	159	.98	150	1.25	141	1.47	134	1.69	128	1.90	122	2.08	117	2.26	113	2.44
	−10	161	.94	152	1.19	144	1.41	137	1.62	131	1.81	126	1.99	121	2.16	117	2.33
7	0	163	.90	154	1.13	147	1.35	140	1.54	135	1.73	130	1.90	125	2.06	121	2.22
	20	167	.81	159	1.02	152	1.21	146	1.39	141	1.55	136	1.70	132	1.85	128	1.98
	30	169	.76	161	.96	155	1.15	149	1.31	144	1.46	139	1.60	135	1.73	132	1.87
	−20	168	.90	159	1.15	151	1.37	144	1.58	138	1.77	133	1.96	128	2.14	123	2.29
	−10	170	.87	161	1.10	153	1.31	147	1.51	141	1.69	136	1.87	131	2.04	126	2.18
8	0	172	.83	164	1.05	156	1.25	150	1.44	144	1.62	139	1.78	134	1.93	129	2.07
	20	175	.75	167	.94	161	1.13	155	1.30	150	1.46	145	1.60	141	1.74	137	1.87
	30	177	.71	169	.89	163	1.07	158	1.23	153	1.38	148	1.51	144	1.64	140	1.76

velocity between 1200 and 1600 feet per minute is permissible. For this purpose velocities are based on an air density corresponding to 70°, this being merely an arbitrary standard adopted for convenience in making computations. In very cold climates a a velocity of 800 feet per minute or less is advisable because of the tendency for the condensation to freeze in the coils. The velocity thus chosen is used both as a basis for computing the height and width of the heater and also for determining its depth, *i.e.*, the number of stacks to be used. In Table XLIX are given the final temperatures obtainable from heaters of vari-

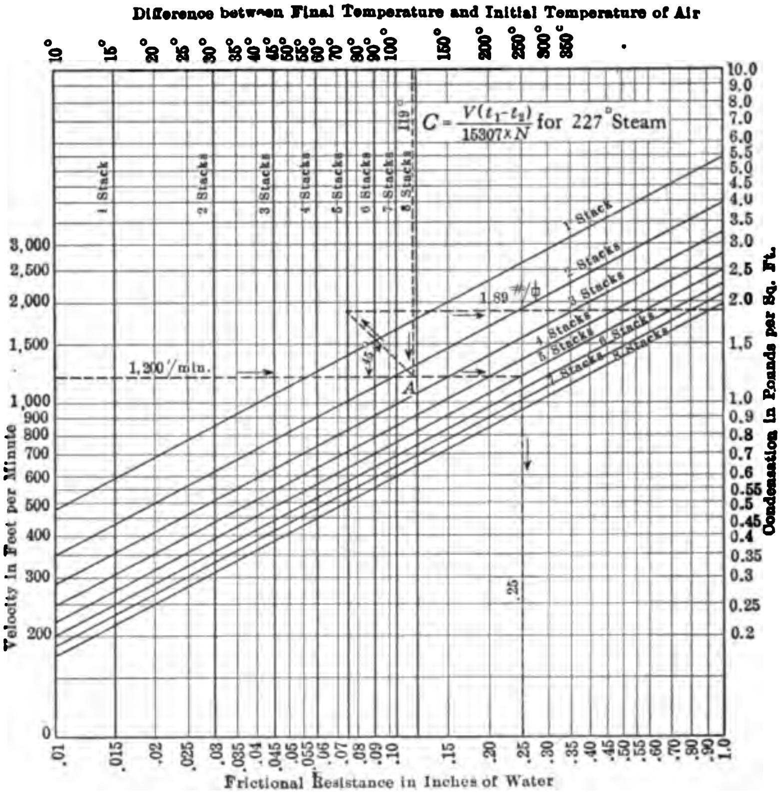

FIG. 152.—Friction curves for pipe coil heaters.

ous depths for air at different initial temperatures and velocities. The final temperature for which the heater is designed depends upon the amount of heat to be supplied and upon whether the fan system is to be used for ventilating alone or to supply the

heating requirements also. The temperature of the entering air used in the computations should be the minimum for which the system is to be designed.

Example.—Assume that a factory is to be heated and that 1,400,000 cubic feet of air per hour are required at a temperature of 140°. Minimum outside temperature 0°. What size Vento heater should be used?

$$\text{Free area (square feet)} = \frac{\text{volume (cubic feet per minute at 70°)}}{\text{velocity (feet per minute)}}$$

$$\text{Free area} = \frac{1{,}400{,}000}{1200 \times 60 \times 1.1320} = 17.17 \text{ square feet}$$

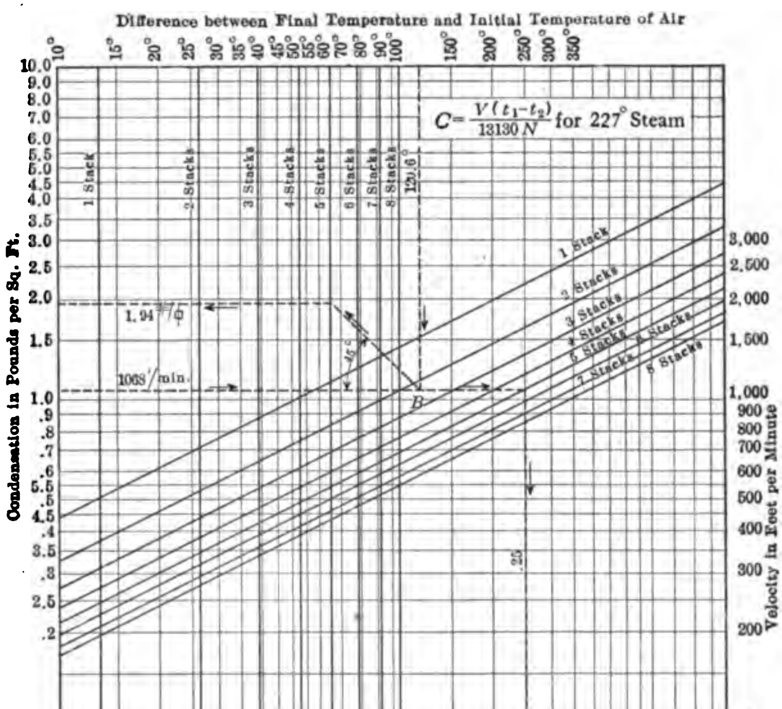

Fig. 153.—Friction curves for vento heaters.

Referring to Table XLVIII it is seen that by using eighteen 60-inch sections, spaced 5 inches center to center, the free area will be 18 × 0.921 = 16.58 square feet, which is sufficient, giving a velocity of 1244 feet per minute. From Table XLIX it is seen that a heater seven stacks deep would raise the air from a temperature of 0° to 140° at a velocity

of 1200 feet per minute. The heater should therefore be sevnn stacks deep. Ordinarily it would be divided into a tempering coil of three stacks and a heating coil of four stacks.

Pipe-coil heaters are chosen in a similar manner from the data furnished by their manufacturers.

Recent tests[1] have shown that the heating effect of both Vento and pipe-coil heaters is closely related to the friction loss undergone by the air in passing through them; and that for the two different types of heaters, the friction loss will be practically identical for the same increase in temperature of the air. This might logically be expected as the heat transmission depends upon the thoroughness of the rubbing action of the air over the heating surfaces.

From the curves in Figs. 152 and 153 the friction drop can be determined for either Vento or pipe coil if the other facts are known, and *vice versa.* These curves are based on the following formula which was developed from the results of tests mentioned above on pipe coils and upon tests made on Vento heaters by L. C. Soule.

$$C = \frac{V(t_1 - t_2)}{KN}$$

in which C = condensation in heater—pounds per square foot per hour.

V = velocity of air—feet per minute.

$t_1 - t_2$ = temperature rise of air.

N = number of stacks in heater.

K = a constant = 15,307 for pipe coil and 13,130 for Vento.

As an example of the use of the charts we will take an assumed case. With five stacks and an entering temperature of 10°, the final temperature for 1200 feet velocity is found from pipe-coil data to be 129°, making the increase in temperature 119°. In Fig. 152 the horizontal dotted line representing 1200 feet velocity intersects the vertical line representing 119° at the point A. From point A we draw the 45° line until it intersects the vertical line for five stacks. From this point we extend a horizontal line to the right-hand side of the chart and we see that the

[1] See "Comparison of Pipe Coils and Cast-iron Sections for Warming Air," by JOHN R. ALLEN, *Proc.* A. S. H. & V. E., 1917.

condensation per square foot per hour is 1.89 pounds. The frictional resistance is obtained by extending the horizontal line for 1200 feet velocity to the right until it intersects the diagonal line for five stacks; a vertical line from this intersection shows the resistance to be 0.25 inches of water. In Fig. 153 the same case is worked out for Vento heaters as indicated by the dotted lines. The condensation is found to be about 1.94 pounds and the velocity 1068 feet for the same resistance and temperature rise. It will be noted that while the heating effect and resistance of the two heaters are the same, the velocities are quite different.

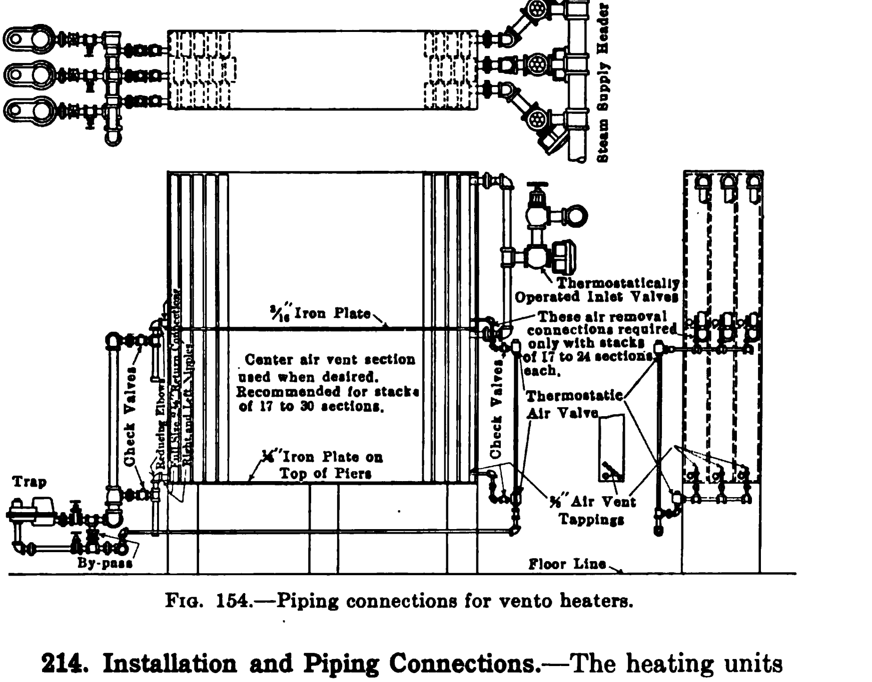

FIG. 154.—Piping connections for vento heaters.

214. Installation and Piping Connections.—The heating units are usually mounted on a brick or concrete pier and enclosed by a metal duct. The proper arrangement of the steam piping connections for Vento heaters is shown in Fig. 154 for a double-tier installation. The center section of a long stack is tapped for an air vent as shown. Separate valves should be provided for each stack or pair of stacks.

Special care is necessary in arranging the return connections from fan heaters, as any accumulation of condensation will soon

be frozen by the cold air. There is always a considerable drop in pressure through the heaters and the inlet connections, so that the pressure at the return connections should not be depended upon to lift the condensation; the discharge should be by gravity or a vacuum pump should be used.

Thermostatic control is almost essential on fan systems. The diaphragm control valves, similar to those used for radiators, are installed as shown in Fig. 154 and are controlled from thermostats whose expansion member projects into the ducts.

Problems

1. In the example in Par. 189, assuming that 657,000 cubic feet of air per hour are delivered, if the heat loss as given was computed for 0°, what should be the delivery temperature when the outside temperature is 20°?

2. A factory building is to be heated by a hot-blast system with complete recirculation. With the following data given compute the amount of air which must be handled per hour by the system.

Heat loss from building	27,800 B.t.u. per hour per degree difference in temperature.
Inside temperature	65°
Outside temperature	−10°
Temperature at which air is delivered.	140°

3. In the single duct system of Fig. 140 assume that the longest duct is to carry 1100 c.f.m. What is the total pressure required in the plenum chamber?

4. Compute the pipe sizes for a trunk duct system similar to that in Fig. 141 except that the air quantities in the different sections on a 70° basis are as follows:

Section	Air quantity
A—B	19,000 c.f.m.
B—C	7,500
C—D	2,000
B—E	6,000
E—F	4,000

Maximum air temperature 130°.

5. Find the speed, horsepower, and outlet velocity for three different sizes of steel plate fan[1] delivering 18,000 c.f.m. against a static resistance of 1½ inches at 70°.

6. Find the speed, horsepower, and outlet velocity for three different sizes of multi-blade fan[1] delivering 12,000 c.f.m. against a static resistance of 2 inches at 70°.

7. A multi-blade fan is to handle 9000 c.f.m. against a static head of 1¼ inches at 140°. What is the required speed and horsepower?

[1] See tables in Appendix, pages 276 to 299.

8. What would be the size of vento heater required to heat 800,000 cubic feet of air per hour from an outside temperature of 10° to a delivery temperature of 140°? Assume a velocity through the heater of 1500 feet per minute.

9. What would be the size of vento heater required to heat 1,100,000 cubic feet of air per hour from an outside temperature of 0° to a delivery temperature of 70°? Assume a velocity through the heater of 1100 feet per minute.

10. Find by means of the friction chart in Fig. 153 the frictional resistance of a vento heater, 5 stacks deep, for a velocity of 1500 feet per minute. Find the resistance of a vento heater, 3 stacks deep, for a velocity of 900 feet per minute.

CHAPTER XVI

AIR WASHERS AND AIR CONDITIONING

215. The Air Washer.—Various methods of filtering or washing air have been in use for many years. In the older forms of apparatus the dust was usually filtered from the air by means of muslin screens; but this method is not very effective and has the disadvantage that the screens soon become clogged with dirt, greatly increasing the resistance to the flow of air through them. Screen filters have been superseded by the modern air washer, in which the dirt is removed from the air by water sprays and by the contact of the air against wet surfaces.

A typical air washer is shown in Fig. 155. It consists of three elements—the spray nozzles, the scrubber plates, and the eliminator plates. The nozzles are placed in a bank across the path of the air and the water is forced through them by a pump and is discharged in a fine conical spray or mist in the direction of the air flow. The air, drawn through the washer by the fan, is thus brought into intimate contact with the water and much of the dirt and soluble gases are removed. The final cleansing is done by the scrubber plates which are designed to change the direction of flow so that the dirt will be thrown out from the air by its inertia and by the rubbing of the air over the wet surfaces. The plates are kept flooded either by the spray nozzles or by a separate row of nozzles placed above them. Following the scrubber plates are a series of eliminator plates whose function is to remove the entrained water from the air. The lower part of the washer constitutes a tank into which the water falls and from which it is taken by the circulating pump. A float valve admits fresh water as required to replace that evaporated.

Proper provision must be made in an air washer to prevent trouble from the large quantities of dirt which are washed from the air and deposited in the tank. A screen of ample area is necessary on the suction line to the pump to prevent the dirt from being carried into the circulating system, and in some types of washers special devices are necessary to enable the spray

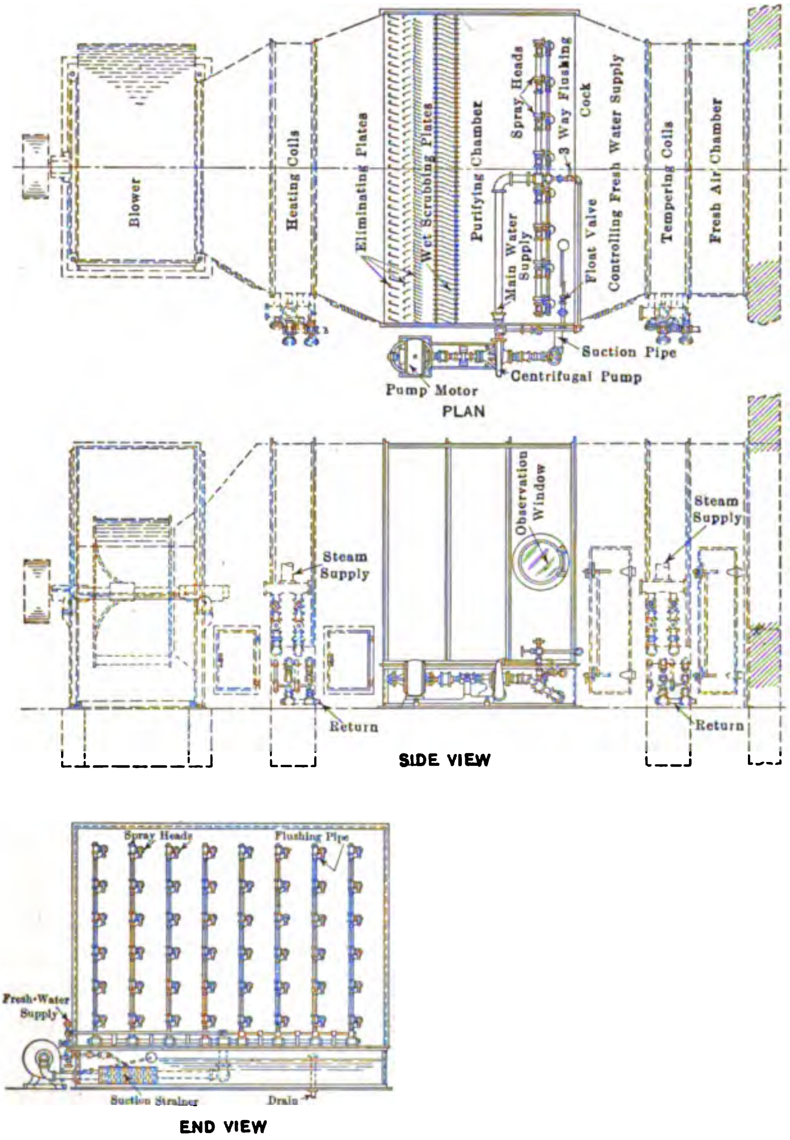

Fig. 155.—Air washer.

nozzles to be cleaned periodically by flushing. The accumulated dirt must be removed from the tank at frequent intervals.

The air washer is placed between the tempering coils and the heating coils of a fan system, this arrangement being necessary in order to insure that the air entering the washer will be at a temperature sufficient to keep the spray water from freezing.

216. Air Conditioning.—The air washer, in addition to cleansing the air, is also used to add to or reduce its moisture content so that the atmosphere in the building will be maintained in accordance with the standard fixed by ventilation or manufacturing requirements. In many textile processes, and in the manufacture of powder, photographic films, etc., the proper "conditioning" of the air is of extreme importance.

Humidification is accomplished by heating the spray water so that the air will absorb the proper amount of moisture while passing through the spray chamber. Sufficient heat is given up by the spray water, first to evaporate sufficient moisture to bring the air to saturation at its entering temperature and, second, to add further amounts of heat and moisture until the air leaves the washer at saturation and at such a temperature that it contains the requisite quantity of water vapor. It then passes to the heating coils which raise its temperature without affecting its moisture content.

For example, suppose that it is required to deliver air to a room at a temperature of 70° and a relative humidity of 60 per cent., which requires a moisture content of 4.85 grains per cubic foot. We will assume that the outside air has a dry-bulb temperature of 25° with a relative humidity of 20 per cent. Referring to Fig. 156, the entering air is heated by the tempering coils to a temperature of 40°, as represented by the line AB. In the washer moisture is absorbed from the spray water until the air becomes saturated at 40° as represented by BC. Both heat and moisture continue to be absorbed from the spray water until the air reaches the condition represented by point D, in which it contains 5.0 grain per cubic foot and has a temperature of 56°. It is then heated by the heating coils to the delivery temperature of 70°, at which it will have the required relative humidity of 60 per cent. During this process the moisture content per pound of air remains the same, the weight of the vapor per cubic foot decreasing slightly because of its expansion due to the temperature increase. For approximate calculations this difference may

be neglected and the line DE representing this last step on the chart in Fig. 156 may be taken as a horizontal line. For very accurate work the charts in Figs. I and II in the Appendix, which are constructed on the basis of 1 *pound* of air, may be used.

Every final condition of the air has a corresponding temperature at saturation, to which the air is brought before it passes to the heating coils. If, in the case given above, the temperature of the outside air were above 56° it would be lowered because of the heat given up by it to evaporate the moisture which it absorbs—

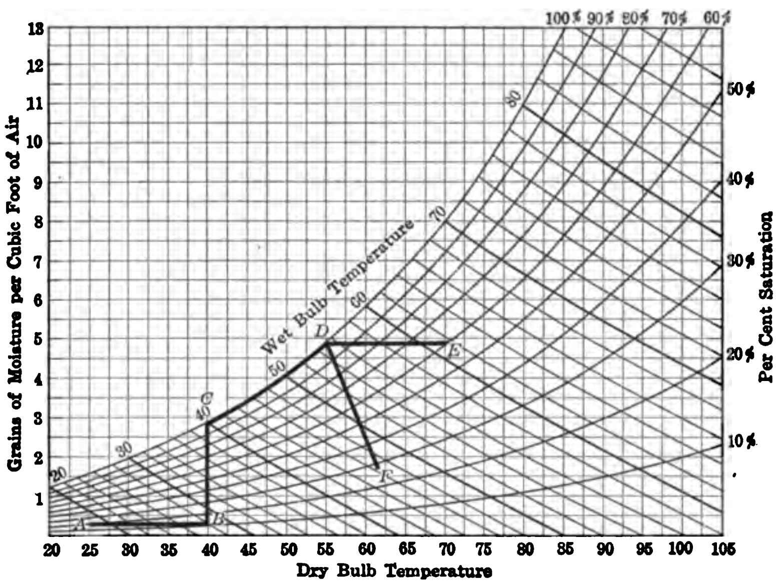

FIG. 156.

provided, however, that its original moisture content be considerably below saturation. The action would then be represented by the line FD. If the dry-bulb temperature of the entering air were between 40° and 56° no heat would be added by the tempering coil and moisture would be added at a constant dry-bulb temperature until the air reached saturation, after which it would follow the line CD to 56° as before.

217. Spray-water Heater.—In order to supply heat to the spray water, a heater is installed in the water circulating line, usually between the pump and the spray nozzles. If high-pressure steam is available it is injected directly into the water

through a suitable valve. If low-pressure steam or hot water are used a closed heater, in which the spray water circulates through tubes surrounded by the heating medium, is necessary.

218. Humidity Control.—The steam supply valve of the heater is controlled—usually by automatic means—so that the proper amount of heat is added to the water. In a compressed-air system of control, a diaphragm valve is placed on the supply to the water heater and may be operated by means of a "hygrostat" or "humidostat," which corresponds to the thermostat of a temperature control system. In place of the thermostatic element there is used some material such as wood or hair which undergoes a change in length when the moisture content of the surrounding air changes. The "humidostat" is placed either in

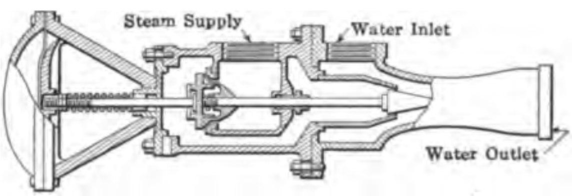

Fig. 157.—Spray-water heater.

the main duct or in the principal room of the building and controls the supply valve on the heater. An injector type of heater with a diaphragm control valve is shown in Fig. 157. Another and a more rational method of humidity control is based on the fact that the air always leaves the washer in a saturated condition and therefore its moisture content will depend upon its temperature. From a thermostat placed in the path of the air leaving the washer the heat added to the spray water is controlled so that the exit temperature of the saturated air is at the point fixed by the humidity required. In the example given in Paragraph 216 the thermostat at the washer outlet would be set for 56° and the temperature of the air leaving the washer would be maintained at that point. A special duct-type thermostat of the form shown in Fig. 158 is used for the purpose, having a bulb extending into the path of the air and controlling the air supply to the diaphragm valve in the usual manner. Humidification may also be accomplished by steam jets when no washer is used, in which case the jets are located in the same position as the washer and may be automatically controlled.

Another type of humidifier is located directly in the room and discharges a finely atomized spray which vaporizes after leaving the apparatus. If the steam supply is perfectly free from oil and does not possess a disagreeable odor, humidifiers of the type which discharge steam directly into the room may be employed. They are not always suitable for use in moderate weather, however, as a considerable amount of heat is given up by the steam which might raise the room temperature to an uncomfortable point. The objection to these latter forms of humidifier is the absence of automatic means of regulating the humidity.

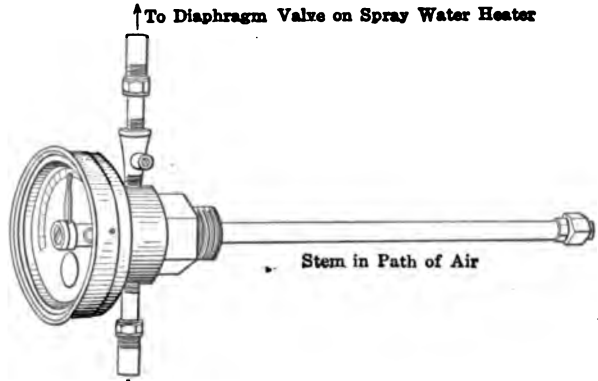

FIG. 158.—Duct thermostat for dewpoint method of humidity control.

219. Cooling and Dehumidification.—If no heat is added to the spray water of an air washer some evaporation will still take place but the latent heat of the vaporization in this case is taken from the air itself. It is by the application of this principle that cooling by means of an air washer is accomplished, the temperature of the air being lowered because of the heat supplied to vaporize the added moisture. The extent of the cooling effect depends upon the capacity of the entering air for absorbing moisture or, in other words, upon the wet-bulb depression of the entering air. As the air absorbs moisture in the spray chamber its dry-bulb temperature drops but the wet-bulb temperature, which is a measure of the total heat of the mixture, remains unchanged. If the water is re-circulated its temperature soon drops to the wet-bulb temperature. In a perfect washer the dry-bulb temperature of the air would be reduced to the same point—*i.e.*,

the air would become saturated, but in a commercial washer this limit is never reached. The cooling effect actually obtained averages about 60 per cent. of the wet-bulb depression, this percentage being termed the **humidifying efficiency** of the washer. Referring to the psychrometric chart in Fig. 159, the point *A* represents the original condition of the air at 90° dry-bulb temperature and 75° wet-bulb temperature. The cooling and humidifying action is represented by the constant wet-bulb temperature line *AB*, the point *B* representing the final condition of 81° dry-bulb temperature. The line *AC* represents the ac-

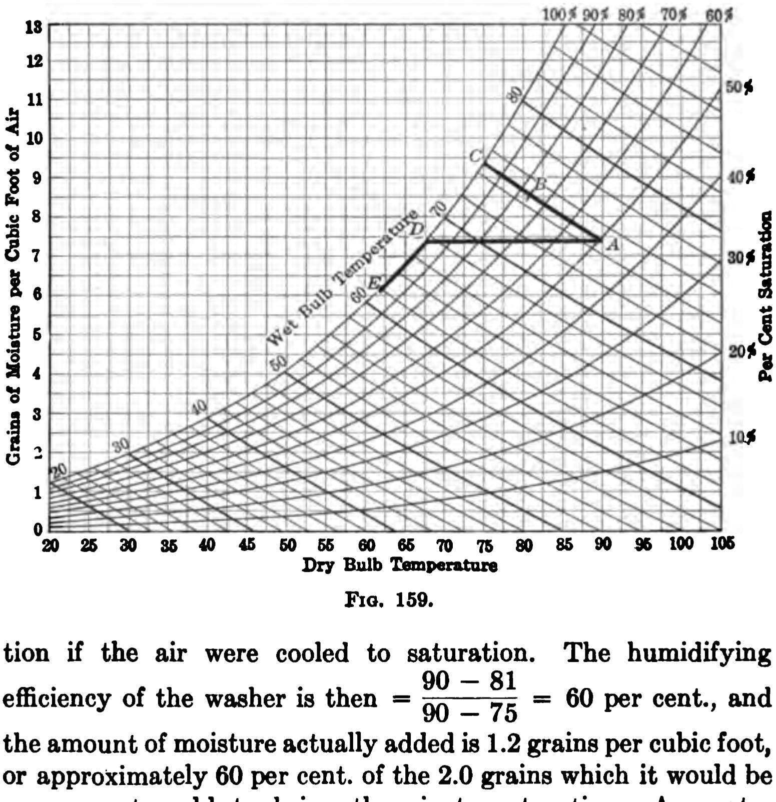

FIG. 159.

tion if the air were cooled to saturation. The humidifying efficiency of the washer is then $= \dfrac{90 - 81}{90 - 75} =$ 60 per cent., and the amount of moisture actually added is 1.2 grains per cubic foot, or approximately 60 per cent. of the 2.0 grains which it would be necessary to add to bring the air to saturation. A greater cooling effect can be obtained if the spray water be artificially cooled, in which case heat will be transferred from the air to water by direct contact and no evaporation will take place. Both the dry-bulb and the wet-bulb temperatures will fall until

they coincide at the dew point. If the spray-water temperature is sufficiently low they will be reduced still further and some of the moisture will be given up by the air. This action is represented by the line *ADE* in Fig. 159. In a properly designed washer the air can be cooled to within a few degrees of the average water temperature. This method of dehumidification is sometimes employed in industrial work.

The cooling of the spray water is usually accomplished by means of a refrigeration plant. The brine coils are placed in the tank of the washer so that the spray water during its cycle passes over them. If a supply of cold artesian well water is available the cost of installation and operation is greatly reduced.

CHAPTER XVII

FAN SYSTEMS FOR VARIOUS TYPES OF BUILDINGS

220. School Buildings.—In school buildings and in various other public buildings, the fan system may be designed to furnish both the heating and ventilating requirements, or may be used to furnish ventilation only, the heating being done by direct radiation. In the former case, owing to the necessity for adjusting the temperature of the air supplied to each individual room, a single-duct system is necessary. When ventilation only is supplied a trunk-duct system may be used as the air is supplied continually at a temperature of about 70°. The arrangement

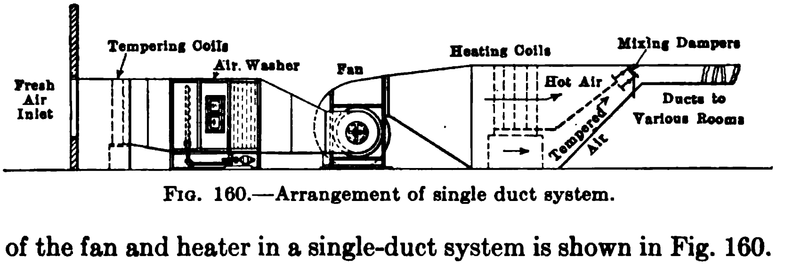

FIG. 160.—Arrangement of single duct system.

of the fan and heater in a single-duct system is shown in Fig. 160. The air passes first through the tempering coils, then through the air washer, if one is installed, and to the fan, which forces part of it through the reheating coils into the hot-air chamber and part of it into the tempered-air chamber. The double dampers at the entrance to each duct are controlled from thermostats in the respective rooms so that the temperature of the mixture of hot and tempered air is sufficient to supply the heat losses from the room and to maintain it at the proper temperature. The volume of air delivered is approximately constant regardless of the relative proportions of hot and tempered air. A mixing damper is illustrated in Fig. 161. The temperature of the tempered air is maintained at about 70° and that of the heated air at about 140°. Sometimes this arrangement is varied slightly by running double ducts to the foot of each vertical duct and installing a mixing damper at that point which is controlled by hand through a chain or cable from the room above.

Provision must be made for removing the air from the rooms at the same rate at which it is supplied and a system of vent flues is provided for that purpose. The flues from the separate rooms join together in a trunk duct and lead to a common discharge at the roof. The attic is sometimes used as a discharge chamber, the flues leading directly to it. Exhaust flues are figured at a velocity of 600 to 750 feet per minute and are assumed to carry off the same amount of air as is delivered to the room. In some cases an exhaust fan is installed to facilitate the removal of the foul air. The velocity in the exhaust flues can then be from 1200 to 1500 feet per minute. In public buildings over three or four stories in height, where the friction in the exhaust flues is appreciable, an exhaust fan is desirable. The ventilation of school rooms is usually done by the downward system, the air entering near the ceiling and being exhausted near the floor.

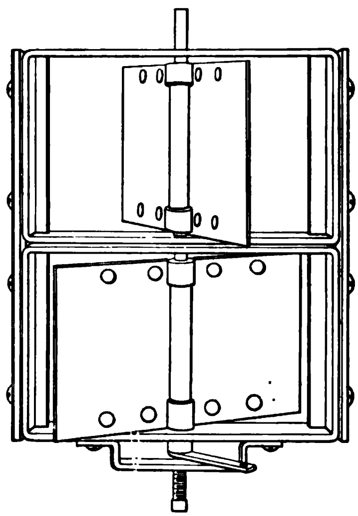

Fig. 161.—Mixing damper.

221. Factory Heating.—The hot-blast system is often the best system for most industrial buildings as it affords a means of supplying fresh air to replace that containing the fumes or moisture from manufacturing processes. It is also desirable in factory buildings where the space required by direct radiation cannot be spared. Owing to the fact that such buildings are seldom divided into many rooms the air can be supplied at a constant temperature through a trunk system of ducts. A draw-through arrangement is almost universally used, the heating coils being placed on the suction side of the fan, which discharges directly into the main duct. For ordinary shop buildings of steel construction, the ducts are of galvanized iron and are suspended from the columns or roof trusses. An example of this arrangement is shown in Fig. 162. In modern reinforced-concrete buildings the columns are frequently made hollow and used as the air ducts, the heating apparatus and the trunk duct being located on the roof and ar-

ranged to discharge the air into the top of each column. Discharge openings are made in the columns at each floor. The

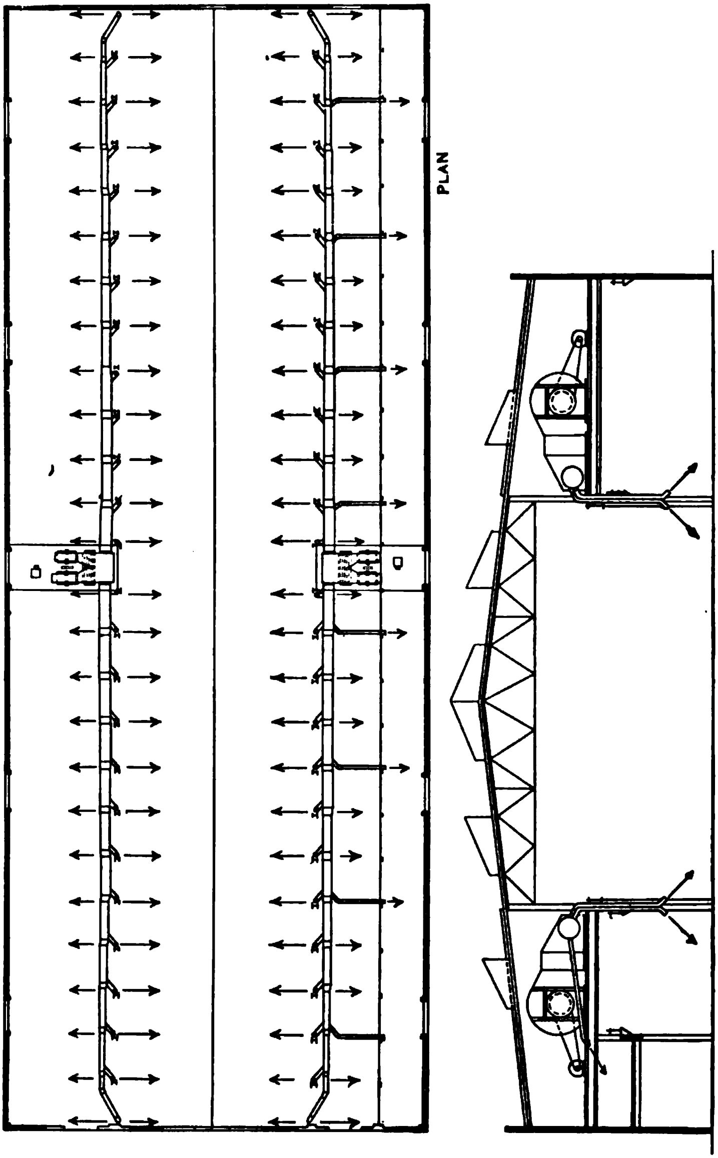

FIG. 162.—Factory hot-blast system.

trunk duct and branch ducts which are on the roof must be well insulated. Details of this method of construction are shown in

Fig. 163. The air is sometimes carried underground in brick or concrete ducts, but the heat loss from such ducts is considerable.

222. Heating of Theatres and Auditoriums.—Theatres and auditoriums are usually both heated and ventilated by the fan system. In a theatre the air is usually admitted through openings in the floor, the space beneath the floor acting as a plenum chamber as shown in Fig. 164. These openings are made adjustable so that the distribution of air throughout the house can be controlled. The foul air is removed through registers near the roof and beneath the galleries. An exhaust fan is often provided. If it is not possible to introduce the air through the floor, registers in the side walls are provided for the purpose. The former system provides a much more even distribution, however.

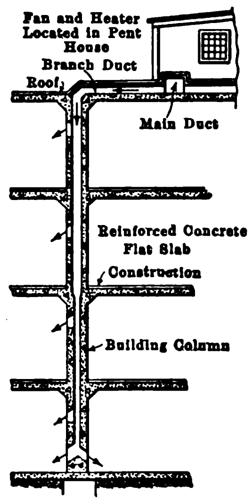

FIG. 163.—Hollow column method of distribution.

Air washers are installed in all first-class theatres, both to remove dust from the entering air and to cool it. Direct radiation is usually necessary in the lobby, offices, and dressing rooms.

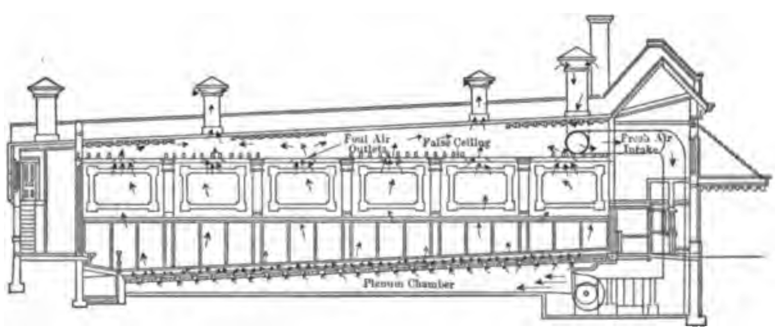

FIG. 164.—Theatre ventilating system.

223. Methods of Estimating Heating Requirements.—It is frequently necessary to estimate the cost of heating a building prior to its construction. It is a very difficult matter to do this accurately, first, because of the inaccuracies that are inevitable in the computation of the heat losses and, secondly, because of

the pronounced effect of the manner in which the firing is done and in which the system is handled.

The most satisfactory method is to compute the theoretical heat loss and to apply a factor to allow for the manner in which it is believed the plant will be handled. To compute the total heat loss from the building, it is necessary to assume the temperature at which the building is to be carried and the average outdoor temperature. The heat required for ventilation will depend upon the amount of air used and the number of hours of use.

Example.—Given a school building heated with direct radiation and equipped with a ventilating system. With the following data furnished, what would be the annual fuel cost?

Heat loss from the building per hour per degree difference in temperature between the inside and outside, 12,500 B.t.u., not including ventilation.
Average outside temperature for heating season, 38°.
Hours use of building, 8:00 a. m. to 4:00 p. m., 5 days per week.
Amount of air supplied for ventilating, 40,000 cubic feet per minute.
Cubic feet of space, 300,000.

The actual time during which the building is used is 8 hours per day. Let us assume that a temperature of 68° is maintained for 10 hours of each of the 5 school days or 50 hours per week. Allowing for vacations, we may assume that the school is occupied for 32 weeks of the heating season, or 1600 hours per year. For the remainder of the 8 months or 5760 hours in the heating season, the temperature may be assumed to average 50°. The heat loss, not including ventilation, would then be as follows:

$$
\begin{aligned}
12{,}500 \times (68 - 38) \times 1600 &= 600{,}000{,}000 \text{ B.t.u.}\\
12{,}500 \times (50 - 38) \times 4160 &= \underline{623{,}000{,}000} \text{ B.t.u.}\\
&\ 1{,}223{,}000{,}000 \text{ B.t.u.}
\end{aligned}
$$

The ventilating fan, if properly handled, would be operated only during the actual hours of occupancy or 40 hours per week, 1280 hours per year. The heat loss from this source would be

$$60 \times 40{,}000 \times 1280 \times 0.019(68 - 38) = 1{,}750{,}000{,}000 \text{ B.t.u.}$$

During the remainder of the time, the air may be assumed to change $1\frac{1}{2}$ times per hour due to infiltration.

$$300{,}000 \times 1.5 \times 4480 \times 0.019(50 - 38) = 460{,}000{,}000 \text{ B.t.u.}$$

The total heat loss is then 3,433,000,000 B.t.u.

Assume that the coal used contains 13,000 B.t.u and costs $5 per ton. For a plant of this nature, operated by efficient help, we may safely assume that 65 per cent. of the heat in the fuel is delivered to the building. The total annual cost would then be

$$\frac{3{,}433{,}000{,}000}{13{,}000 \times 0.65} \times \frac{5}{2000} = \$1015$$

This is the estimated cost on a strict basis. It would be well to add about 10 per cent. for safety, making the final estimate $1116.50. If unskilled help were to have been used or other known factors tending to extravagance in the use of heat, it might be necessary to increase the strict figure by as much as 30 per cent. in extreme cases.

224. Heating Requirements of Various Types of Buildings.—The variation in the amount of heat used in different types of buildings is shown in Table L, which gives data for a number of steam-heated buildings in Detroit, Michigan. These buildings are all heated from a central station. The heat loss per hour per degree difference in temperature is given for each building. It will be noticed that the steam consumption per B.t.u. of computed heat loss varies greatly for the individual buildings and that the average figures for the different classes of buildings are also quite different.

TABLE L.—STEAM CONSUMPTION OF BUILDINGS AT DETROIT, MICHIGAN
Heating Season of 1914–15
Average Temperature for Heating Season (Oct. 1 to May 31)—38.9°

	Installed radiation, square ft.	Cubic contents, cubic ft.	Computed heat loss[1]	Steam consumption for heating season[2]	Steam consumption per square ft. of installed radiation (Col. 4 ÷ Col. 1)	Steam consumption per thousand cubic ft. of space (Col. 4 ÷ Col. 2)	Steam consumption per B.t.u. of computed heat loss (Col. 4 ÷ Col. 3)
OFFICE BUILDINGS							
Building No.							
1	6,524	549,000	26,600	3,091,264	474	5,630	116.2
2	2,755	326,000	16,000	2,393,000	868	7,330	149.5
3	3,820	273,000	13,100	1,860,676	487	6,810	142.0
4	5,280	367,000	16,700	3,563,200	668	9,700	213.5
5	15,300	1,350,000	65,000	12,632,048	825	9,350	194.2
6	7,940	584,000	29,100	4,942,767	622	8,460	169.8
7	50,000[3]	3,220,000	120,000	34,209,387	684	10,630	285.0
8	79,500[3]	4,900,000	205,000	41,850,000	527	8,540	204.2
Totals and weighted averages	171,119	11,569,000	491,500	104,542,342	**610**	**9,020**	**212.5**
RETAIL STORE BUILDINGS							
Building No.							
1	1,673	160,960	8,715	627,200	375	3,900	71.9
2	1,256	111,500	6,400	364,700	290	3,270	57.0
3	16,100[3]	2,725,100	104,000	7,254,078	451	2,660	69.8
4	11,315[3]	1,063,100	42,400	6,012,348	531	5,660	141.9
5	3,864	403,000	18,700	2,110,900	550	5,250	112.9
6	2,684	459,400	18,400	987,000	368	2,150	53.6
7	4,413	325,500	17,700	1,677,800	380	5,140	94.6
8	1,701	199,000	8,690	1,437,600	843	7,210	165.0
9	3,632	613,000	21,600	3,133,650	862	5,110	145.0
10	2,620	393,000	16,500	1,539,560	587	3,910	93.2
11	2,513	350,000	11,890	2,214,200	880	6,320	186.1
12	2,162	197,800	8,200	1,072,900	496	5,420	130.8
Totals and weighted averages	53,933	7,001,360	283,195	28,431,936	**527**	**4,060**	**100.5**
RESIDENCES: Totals and averages for 114 buildings	65,421	3,156,800	304,499	37,484,000	**573**	**11,870**	**123.0**
GARAGES: Totals and averages for 12 buildings	11,414	1,219,700	74,243	9,949,800	**870**	**8,160**	**134.0**

[1] B.t.u. per hour per degree difference between inside and outside temperatures.
[2] Including steam for heating water.
[3] Including equivalent of fan coil.

CHAPTER XVIII

CENTRAL HEATING

225. Location of Power Plant.—It is not intended in this chapter to discuss the design of heating systems, such as are installed for the purpose of heating parts of a city, but rather to describe the methods used in distributing heat to groups of buildings such as public institutions; and as the conditions under which different systems are installed differ widely, the suggestions which follow can be but general.

Before starting the design of the distribution system it is first necessary to have a careful survey made of the property, showing the location of the buildings to be heated and the elevation of their basements and first floors, together with a general profile of the ground through which the tunnels or pipes are to be run. The profile of the ground will largely determine the proper location of the power house. In general, the power house should be located as nearly as possible to the buildings to be heated or as nearly as possible to the largest steam load, but the facilities for receiving coal should also be taken into consideration. If it is possible to locate the plant on a siding from which coal can be delivered direct from the cars to the bunkers without trucking, this will often prove to be the most economical arrangement even if it necessitates locating the plant at some distance from the buildings to be heated; for the cost of loading, trucking, and unloading will usually overbalance the investment charges on the additional length of steam pipes required if the plant is located in the more distant location.

If possible the plant should be so located that the condensation from the various buildings can be drained to it by gravity, and it should also be located so that the floor of the boiler room can be drained to the sewer. Considerable difficulty is usually experienced in carrying away the water from the cleaning and blowing down of the boilers if no sewer connection can be made. The question of the soil, the water supply, and the general appearance of the power house must also be taken into consideration.

226. Boilers.—The selection of boilers of the proper type and size is of extreme importance in the economical operation of the plant. A thorough study should be made of the heating and electric load, both present and future. The maximum demand for steam for heating should be computed on a basis of the radiation installed plus a liberal allowance for transmission losses. The demand for steam due to the lighting and power requirements should be computed from a knowledge of the maximum current demand and the steam consumption of the electric generating units, allowing also for the energy used by the power-plant auxiliaries. The boiler capacity must be such as to fill whichever of the two requirements proves to be the greater. The exhaust steam should always be utilized insofar as possible for heating. When the available exhaust is not sufficient, some live steam must be used, while if there is more exhaust steam than can be utilized some of it must be discharged to atmosphere.

After having determined the maximum amount of steam which the plant might be called upon to furnish, the size of the boilers can be chosen. The steam output per rated boiler horsepower varies considerably according to the type of boiler, type of furnace, etc., but a rough rule for small plants is to assume that 1 square foot of heating surface in a boiler will evaporate 3 pounds of water per hour. The total boiler capacity can then be computed upon this basis and it should be divided into units of such sizes that the expected range of loads can be handled by operating the boilers within their range of highest economy. This can best be done by providing a certain boiler or boilers to handle the lightest loads which are expected and other boilers to handle the average operating load and the maximum load. It is desirable that there be a sufficient number of boilers in the plant so that the largest one can be cut out of service at any time for cleaning or repairs.

If the boiler pressure to be carried is less than 100 pounds, either fire-tube or water-tube boilers may be used. In general, for this service fire-tube boilers are very satisfactory, as they have large water storage, repairs are easily made, and the boiler may be operated at an output considerably beyond its rated capacity.

The principal objection to fire-tube boilers, except those of the Scotch marine type, is the large space which they occupy. If the boilers are to be operated at pressures much over 100

pounds as will usually be the case if electric generating units are installed, then only water-tube or Scotch marine boilers should be used.

227. Systems of Distribution—Gravity System.—The common method of distributing heat is to pipe the steam to the various buildings and return the condensation to the power house. If the elevation of the power plant with respect to the other buildings will permit, the condensation may be returned by gravity to the boiler and no pumping is necessary. With this system any difference in steam pressure between the boiler and the extreme point in the piping system will result in a corresponding elevation of the water level in the return system at the extreme point—each pound of pressure difference producing a difference in level of 2.31 feet. It is essential, then, that with a gravity-return system the difference in pressure between the boiler and the extreme point of the piping system be comparatively small. The drop in pressure assumed will determine the size of the steam piping. In gravity systems it is usual to allow for a drop in pressure of not over 2 pounds between the boiler and the extreme end of the system. In some cases the gravity-return system has been used over quite an extended area, one building so heated being as far as 2500 feet from the boiler, and the system has given very good satisfaction.

In a central heating plant using the gravity-return system, unless the steam mains are from 6 to 8 feet above the return pipes, it is necessary that the steam condensed in the mains be dripped into a separate return line and pumped back to the boilers, by a pump or a return trap. The pump or trap should be of sufficient size to take care of the large amount of condensation which occurs when the steam is first admitted to the cold pipes. By returning the condensation of the mains separately, hammering is avoided and the system can be started much more rapidly.

Gravity-return systems are rarely used where the boiler pressure exceeds 10 pounds.

228. Low-pressure Pump Return System.—In a very large system where it is difficult to get enough difference in elevation between the steam and return mains, or where the drop in pressure exceeds 2 pounds, it is usual to install a pump return system. This will usually be necessary in case any of the buildings are piped with a two-pipe vapor system as the difference in

pressure between the main and return is then quite liable to be over 2 pounds. One of the common arrangements is to discharge the condensation from each building through a trap into the return main which carries the water back to a tank in the power house. From this tank the water is returned to the boilers by means of a pump. The drip from the steam main is trapped directly to the return main.

229. High-pressure System.—Steam is sometimes distributed at high pressure and the pressure reduced before entering the building piping systems by means of a reducing valve. This method has some advantages. Because of the higher pressure, the allowable pressure drop in the distributing pipes is greatly increased. This, together with the fact that the specific volume of the steam is less at the higher pressure, allows the use of much smaller pipes in the distribution system and thereby reduces its cost. In determining the size of the steam mains, a considerable drop may be allowed under maximum conditions, providing the pressure at the most distant building is always sufficient to heat the building.

230. Combination of Power and Heating System.—In the majority of cases the heating system is combined with an electric lighting and power system. The piping connections may be made in a manner quite similar to the arrangement in Fig. 97, page 140, provision being made to feed live steam to the heating mains to supplement the exhaust steam when the latter is less than the heating requirements. A back-pressure valve should be provided to insure against the building up of an excessive pressure in the heating mains. When the heating load is very large in comparison with the electrical load, part of the boilers can be used as high-pressure boilers and the others can be lower priced low-pressure boilers connected directly to the heating lines. The desirability of such an arrangement, however, is determined entirely by local conditions.

231. Hot-water Heating.—A hot-water system, using forced circulation, is very satisfactory if properly designed. The water is heated in a tube heater by the exhaust steam and is circulated through the system by means of a centrifugal pump. A vacuum can be carried on the engine exhaust to a degree depending upon the outgoing temperature of the water. To supplement the exhaust steam heater a live steam heater is installed, but in most cases it need be operated only in the coldest weather. The

temperature of the outgoing water is adjusted by the operating engineer for the prevailing weather conditions in accordance with a prearranged schedule.

The distribution lines in a hot-water system may be arranged according to either of two schemes. In the one-pipe circuit system a single main makes a complete circuit of the territory covered and the supply connection to each building is taken from the top of the pipe and the return connection is made to the bottom of the pipe a few feet further along and a resistance is inserted in the pipe between the connections which has the effect of diverting the water into the building system.

In the multiple or two-pipe system both a flow- and a return-main are installed, the water passing from the flow main through the building systems and back to the plant via the return main. The multiple system is the more commonly used although it is somewhat the more expensive to install.

The systems in the buildings are arranged in the ordinary manner for either system of distribution.

232. Methods of Carrying Pipes.—The pipe lines serving the buildings should always be carried underground if possible.

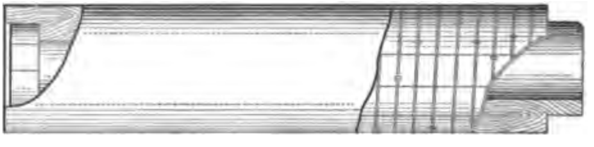

FIG. 165.—Wood casing.

Pipes installed above ground are extremely unsightly and are difficult to support and to insulate. Underground pipes may be installed either in a small conduit or in a tunnel of walking height. The former is a much cheaper method and is quite satisfactory when only one or two pipes are to be installed, but when a greater number of pipe lines must be provided for or when electric cables are also to be installed, a walking tunnel is desirable. There are a large number of designs of conduits ranging from a rough wooden box to a heavily insulated and waterproofed covering. The essential requirements in a conduit for heating pipes are—good insulating qualities, protection of the pipe from water, provision for free expansion of the pipe, and durability.

A very common form of covering is the wood casing shown in Fig. 165. The casing has a wall 4 inches thick and is built of segmental staves bound tightly together with steel or bronze wire, and the assembled casing is rolled in tar and sawdust to give it a waterproof coating and is lined with bright tin to reduce the radiation loss from the pipe. Wood is a very good insulator and if installed under favorable conditions, this form of conduit is very satisfactory. The wood deteriorates, however, if subjected to continued dampness.

The concrete conduit shown in Fig. 166 has the advantage of being very durable and is very easily constructed from common materials. The concrete prevents any considerable amount of

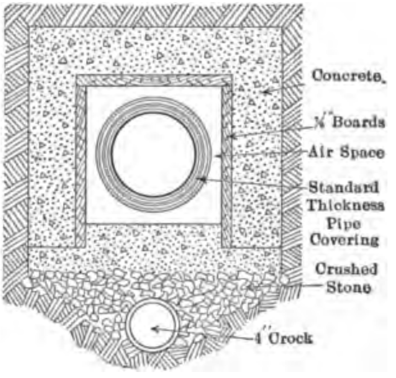

FIG. 166.—Concrete conduit.

water from reaching the pipe and if desired can be made nearly waterproof by the addition of a waterproofing compound. In building this conduit the concrete bottom is first poured and allowed to set and then the pipe is installed and covered with ordinary pipe covering. The wooden box is then built over it and the remainder of the envelope is poured, the sides of the trench serving as the outer sides of the form if the soil is sufficiently cohesive.

The supports for the pipe in any form of conduit must be such as to allow it to move freely when it undergoes a change in length. Some form of roller is commonly used and they are placed at intervals of 10 or 15 feet.

Another form of conduit is built of vitrified tile split longitudinally and having insulating material either molded to the walls

of the tile or packed around the pipe. The joints are cemented to render them water-tight. Such a conduit is shown in Fig. 167. There are many other types of construction in use but those which

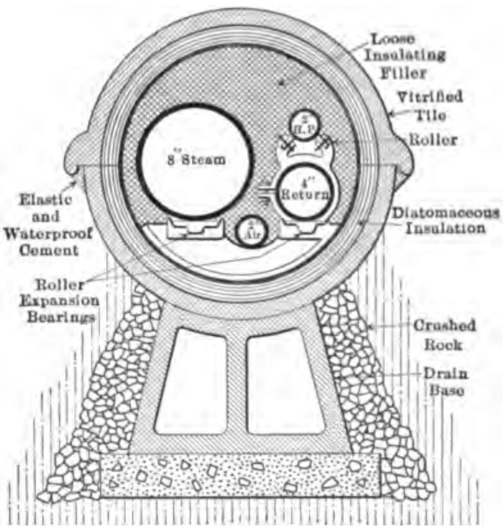

FIG. 167.—Split tile conduit.

have been described are representative. The heat loss from underground lines depends upon the steam temperature, efficiency of the insulation, and the soil conditions. Tests made on the district heating mains of the Detroit Edison Company in 1913–14,

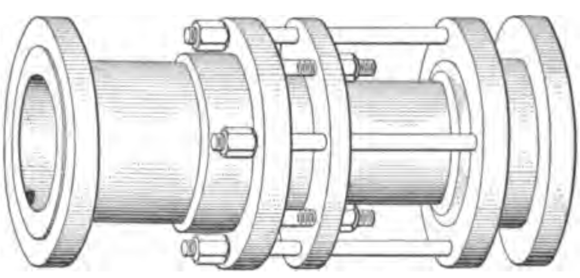

FIG. 168.—Slip joint.

which are laid in conduit of the forms shown in Figs. 165 and 166, gave a result of 0.0511 pounds of condensation per square foot of external pipe surface per hour for steam at 5 pounds pressure.

233. Expansion Fittings.—Owing to the length of the pipe

lines special provision is necessary to take care of the expansion. It is seldom feasible to do so by means of bends, and special fittings are required. The slip joint illustrated in Fig. 168 is a simple means of absorbing large amounts of expansion. It consists of a sleeve which is free to move in the body of the fitting, a packing gland being provided to prevent leakage. Slip joints are located at intervals of from 200 to 300 feet depending upon the steam temperature. They must be installed in manholes or in some other place where they are accessible for packing. The type of expansion fitting shown in Fig. 169 depends upon the

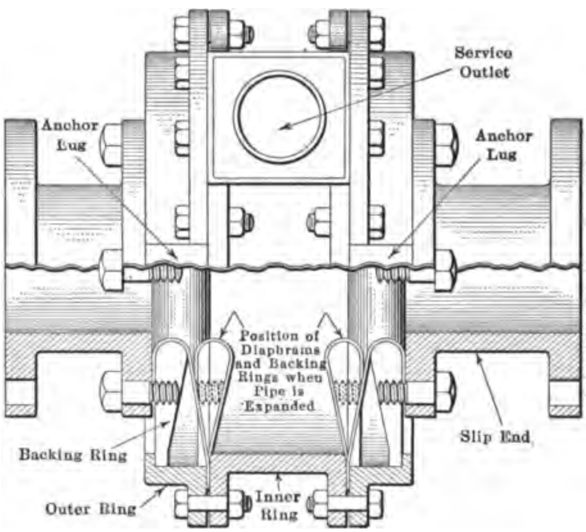

FIG. 169.—Diaphragm expansion joint.

flexibility of a copper diaphragm for absorbing the movement of the pipe. The advantage of such a fitting is that it requires no manhole and does not need to be packed. The amount of travel which can be allowed for each fitting is small, the fittings being usually placed at intervals of 80 to 100 feet and the pipe anchored midway between them. The body of the fitting is also anchored and the expansion of the pipe on either side is taken up by the diaphragms. The cost of a pipe line fitted with diaphragm joints is considerably greater than when slip joints are used.

234. Installation of Underground Lines.—Careful provision should be made for carrying away the ground water from the pipe, particularly if the soil is of clay. A drain tile is installed

for the purpose, either directly below or to one side of the conduit and is surrounded with crushed stone or coarse gravel extending around the lower part of the conduit. Water seeping through to the conduit finds its way into the tile, which carries it away to the sewer. Unless this provision is made, the water will reach the pipe and will corrode it very rapidly.

235. Tunnels.—Tunnels of brick or concrete are used when several pipes are to be carried. The size and shape of tunnel used will depend upon the number of pipes to be carried, the

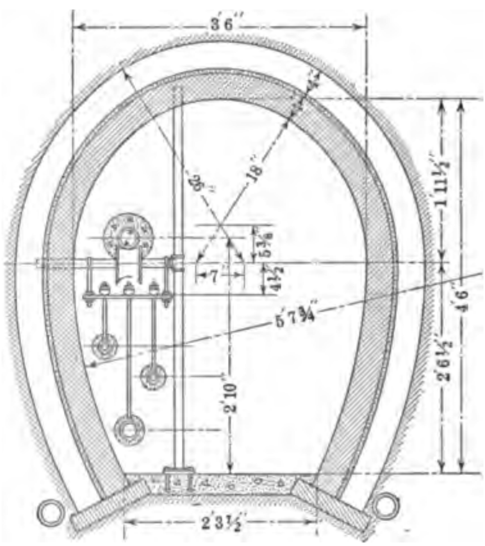

FIG. 170.

character of the soil, and the depth of the tunnel in the ground. Fig. 170 shows a small tunnel suitable for pipes of about 8 inches diameter or less. It is of brick 4 inches thick and has a layer of Portland cement on the outside which is painted with a thick coat of tar or asphalt over the arch to keep out water. Ribs 4 inches thick and 8 inches wide are placed where the supports are imbedded in the walls. The supports are of ordinary pipe. A drain tile may be placed on either side to carry away the ground water but no such provision is necessary if the tunnel is built in a sand or gravel soil. Owing to the small size of this tunnel and its low head room it is not very suitable for large pipes or when much walking through it is necessary.

In Fig. 171 is shown a larger tunnel of the same general shape. It is 6 feet high and 5 feet wide giving ample space for several pipes. In Fig. 172 is shown another form of tunnel of still larger dimensions. The space under the walkway is used for cable ducts. Pipes can be installed on both sides of the tunnel if desired. This shape of tunnel is not suitable for use at considerable depths below the surface because of its flat sides, which offer little resistance against earth pressure. The horseshoe shapes previously described should be used in such cases.

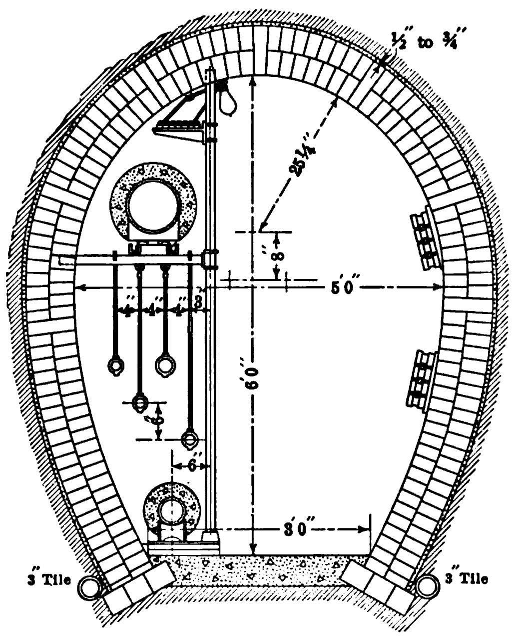

FIG. 171.

236. Size of Pipes.—The size of steam pipes to be used depends upon the amount of steam flowing, the steam pressure, and the available pressure drop. If exhaust steam is used the pressure drop is limited by the allowable back pressure. In general it is necessary to maintain at least 1½ or 2 pounds pressure at each building and in the coldest weather it may be necessary to carry a still higher pressure, especially if the piping in the buildings is not liberally designed. The required pipe sizes can be determined by means of the chart in Fig. 95, page 135, for low-pres-

sure work. In underground piping the noise in the pipes is not a factor and advantage can therefore be taken of all of the available pressure drop to decrease the size of the pipes. In a high-pressure system very much greater pressure drops are permissible and the pressure may be allowed to drop, under maximum conditions, from the boiler pressure nearly to the pressure required for heating. It should be borne in mind, however, that the pressure drop varies as the square of the weight of steam flowing and

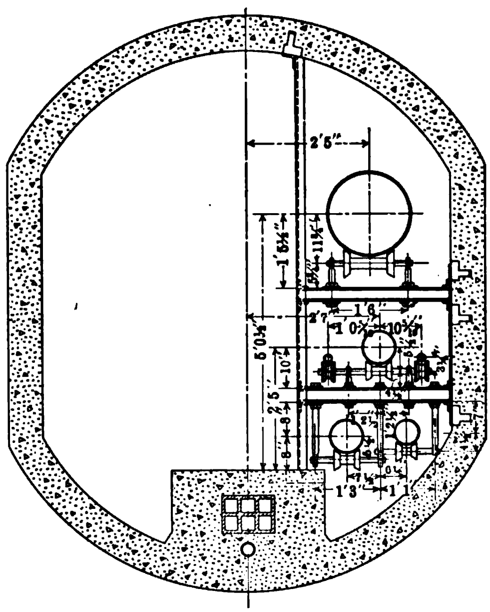

FIG. 172.

consequently a steam flow slightly greater than that estimated will cause a considerably greater pressure drop. It is therefore best to allow a reasonable margin in selecting the pipe sizes. The chart in Fig. 95 is suitable only for pressures of approximately 2 pounds. For higher pressures the capacity of various size pipes for a given pressure drop can be found from the basic formula of Par. 118.

For hot-water systems the pipe sizes can be computed by the methods given in Chapter X.

APPENDIX

TABLE I.—COEFFICIENTS OF HEAT TRANSMISSION THROUGH BUILDING MATERIALS

Walls

Brick Walls

Coefficient of heat transmission, (k) B.t.u. per square foot per hour per degree difference of temperature.

Thickness, inches	Plain	Plastered on one side	Furred and plastered
	k	k	k
4	0.52	0.50	0.28
8½	0.37	0.36	0.23
13	0.29	0.28	0.20
17½	0.25	0.24	0.18
22	0.22	0.21	0.16
26½	0.19	0.18	

Concrete Walls

Thickness, inches	Plain	Furred and plastered	Thickness, inches	Plain	Furred and plastered
	k	k		k	k
2	0.69	....	16	0.37	0.24
4	0.55	0.31	20	0.33	0.23
6	0.49	0.30	24	0.30	0.215
8	0.47	0.28	28	0.27	0.20
10	0.45	0.265	32	0.25	0.18
12	0.43	0.25	36	0.23	0.17

Brick Walls, Sandstone Faces

Thickness of brick, inches	Thickness of sandstone, inches	k	Thickness of brick, inches	Thickness of sandstone, inches	k
4	4	0.31	12	8	0.16
8	4	0.22	4	12	0.26
12	4	0.17	8	12	0.19
4	8	0.29	12	12	0.15
8	8	0.20			

TABLE I.—COEFFICIENTS OF HEAT TRANSMISSION THROUGH BUILDING MATERIALS (*Continued*)

Walls

LIMESTONE WALLS

Thickness, inches	Furred and plastered	Thickness, inches	Furred and plastered
	k		k
12	0.49	28	0.31
16	0.43	32	0.28
20	0.38	36	0.26
24	0.35	40	0.24

TILE WALLS

Thickness, inches	Plain tile	Tile and stucco	Tile, stucco, and plaster
	k	k	k
4	0.79	0.75	0.34
8	0.56	0.54	0.27
12	0.44	0.41	0.26
16	0.40	0.37	0.23
20	0.33	0.31	0.20

WOODEN WALLS

	k
Clapboard 7⁄16 inch, studding, lath and plaster	0.44
Clapboard 7⁄16 inch, paper, studding, lath and plaster	0.31
Clapboard 7⁄16 inch, sheathing ¾ inch, studding, lath and plaster	0.28
Clapboard 7⁄16 inch, paper, sheathing ¾ inch, studding, lath and plaster	0.23

MISCELLANEOUS WOODEN WALLS

Thickness of board, inches	Pine boards only	Double boards, paper between	Board and corrugated iron
	k	k	k
½	0.77	0.32	0.45
1	0.51	0.24	0.36
1½	0.43	0.19	0.30
2	0.35	0.16	0.26
2½	0.30	0.14	0.23

INSIDE PARTITIONS:

	k
Lath and plaster, one side	0.60
Lath and plaster, both sides	0.34

TABLE I.—COEFFICIENTS OF HEAT TRANSMISSION THROUGH BUILDING MATERIALS (*Continued*)

Floors

Floors near ground, assuming ground temperature = 50°

	k
Cement or tile, no wood above	0.31
Cement or tile, wood above	0.08
Dirt floor	0.23
Single thickness wood, on joists	0.10
Double thickness wood, on joists	0.08

Ceilings

	k
Cement or tile, no wood above	0.39
Cement or tile, wood floor above	0.10
Lath and plaster, no floor above	0.32
Lath and plaster, single floor above	0.26
Metal lath and plaster, no floor above	0.49

Roofs

	k
METAL ROOFS:	
Tin on 1-inch sap wood roofing boards	0.45
Copper on 1-inch sap wood roofing boards	0.45
Unlined metal	1.30
Corrugated iron	1.50
Iron over tongue and groove boards	0.20
Iron on wood for framing only	1.32
SLATE ROOFS:	
Unlined slate	0.82
Slate on 1-inch sap wood roofing boards	0.43
Slate over tongue and groove boards	0.30
Slate on wood for framing only	0.80
TILE ROOFS:	
Tile ⅝ to 1 inch thick	0.80
Tile on boards	0.30
MISCELLANEOUS:	
Shingles on narrow 1-inch wood strips	0.33
Tar paper on 1-inch sap wood roofing boards	0.44
Tar and gravel over tongue and groove boards	0.30

Roofs

	k
MISCELLANEOUS (*Continued*):	
Six-inch hollow tile, 2-inch concrete, tar and gravel	0.36
Same, but with 8-inch tile	0.30
Two-inch concrete, with cinder fill	0.80
Four-inch concrete, with cinder fill	0.60
Six-inch concrete, with cinder fill	0.54

TABLE I.—COEFFICIENTS OF HEAT TRANSMISSION THROUGH BUILDING MATERIALS (*Continued*)

Windows, Skylights, and Doors

Average single windows	1.09
Small size windows of ordinary glass	1.20
Single large windows of plate glass	1.08
Double windows	0.45
Single-frame windows with double glass	0.72
Single skylight	1.50
Double skylight	0.50
Single monitor	1.25

Doors

Thickness, inches	Pine	Oak	Thickness, inches	Pine	Oak
	k	k		k	k
½	0.56	0.70	1¼	0.36	0.54
¾	0.47	0.63	1½	0.32	0.50
1	0.41	0.58	2	0.27	0.43

TABLE II.—THERMAL PROPERTIES OF WATER[1]

Temperature, degrees F.	Specific volume, cubic feet per pound	Density, pounds per cubic foot	Specific heat
20	0.01603	62.37	1.0168
30	0.01602	62.42	1.0098
40	0.01602	62.43	1.0045
50	0.01602	62.42	1.0012
60	0.01603	62.37	0.9990
70	0.01605	62.30	0.9977
80	0.01607	62.22	0.9970
90	0.01610	62.11	0.9967
100	0.01613	62.00	0.9967
110	0.01616	61.86	0.9970
120	0.01620	61.71	0.9974
130	0.01625	61.55	0.9979
140	0.01629	61.38	0.9986
150	0.01634	61.20	0.9994
160	0.01639	61.00	1.0002
170	0.01645	60.80	1.0010
180	0.01651	60.58	1.0019
190	0.01657	60.36	1.0029
200	0.01663	60.12	1.0039
210	0.01670	59.88	1.0050
220	0.01677	59.63	1.007
230	0.01684	59.37	1.009
240	0.01692	59.11	1.012
250	0.01700	58.83	1.015

[1] Condensed from Marks and Davis "Steam Tables."

PSYCHROMETRIC CHARTS

The curves in Figs. I and II[1] give the complete properties of air based on the pound of air as a unit. The curves in Fig. I are to be used for dry-bulb temperatures of from 20° to 110° and those in Fig. II for dry-bulb temperatures of from 80° to 380°. Having given the wet- and dry-bulb temperatures of the air, the moisture content in grains per pound of dry air is found by passing vertically from the dry-bulb temperature on the horizontal scale to the diagonal line corresponding to the wet-bulb temperature and thence horizontally to the scale of moisture content at the left. The dew point is determined by passing horizontally to the left from the intersection of the dry-bulb and wet-bulb temperature lines to the saturation curve, the point of intersection being the dew point. The heat required to raise the temperature of 1 pound of air plus its moisture content when saturated, and the corresponding vapor pressure are found by passing vertically from the dew point to the respective curves and thence to the corresponding scales at the left. The total heat is found by passing vertically from the wet-bulb temperature on the saturation curve to the total heat curve and thence to the scale at the left. The volume of air in cubic feet per pound for saturated air and for dry air is obtained by passing vertically from the dry-bulb temperature to the respective curves and to the scale at the left.

Example.—Assume dry-bulb temperature = 75°
relative humidity = 60 per cent.

From the chart we obtain:

Wet-bulb temperature, 65.25°; dew point, 60°; grains moisture per pound dry air, 77; heat required to raise 1 pound air plus its moisture content when saturated at 60° through 1°, 0.247 B.t.u.

Vapor pressure of air saturated at 60°, 0.523 inches mercury. Total heat in 1 pound of air with its moisture content when saturated at 65.25°, 29.75 B.t.u.

As to this last quantity, the total heat of saturated air at 65.25° is the same as that of the air under the given conditions, 65.25° being the wet-bulb temperature.

[1] From "Fan Engineering," Buffalo Forge Company.

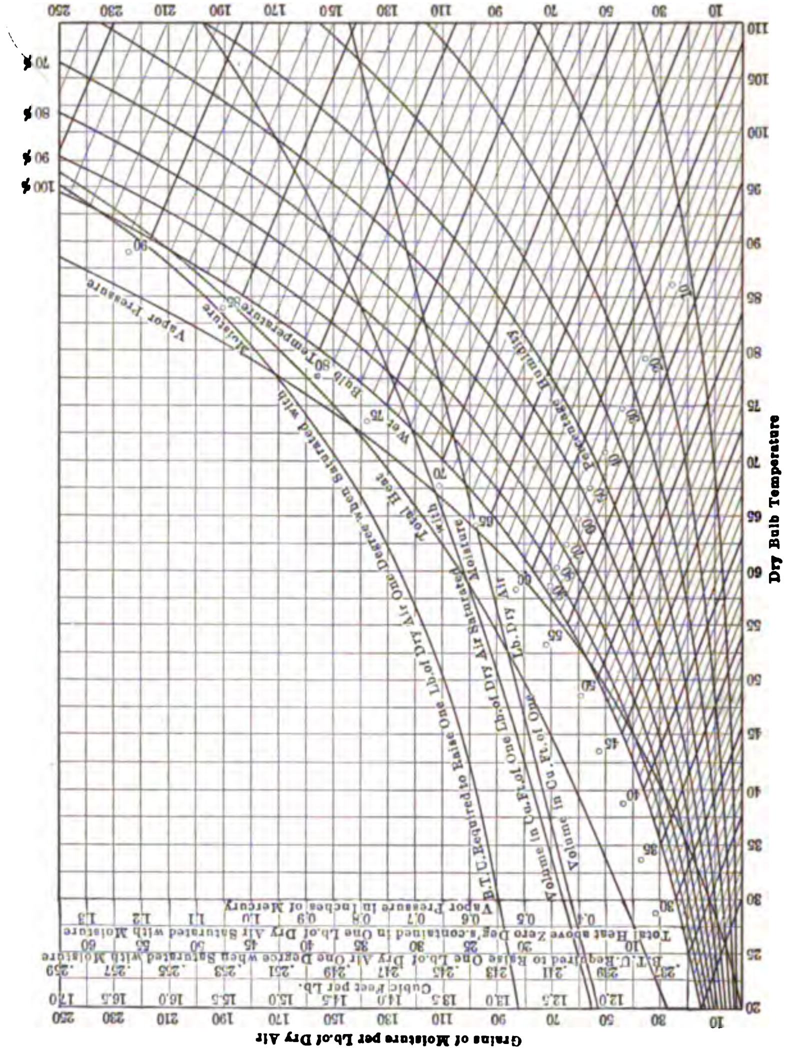

FIG. I.—Psychrometric chart.

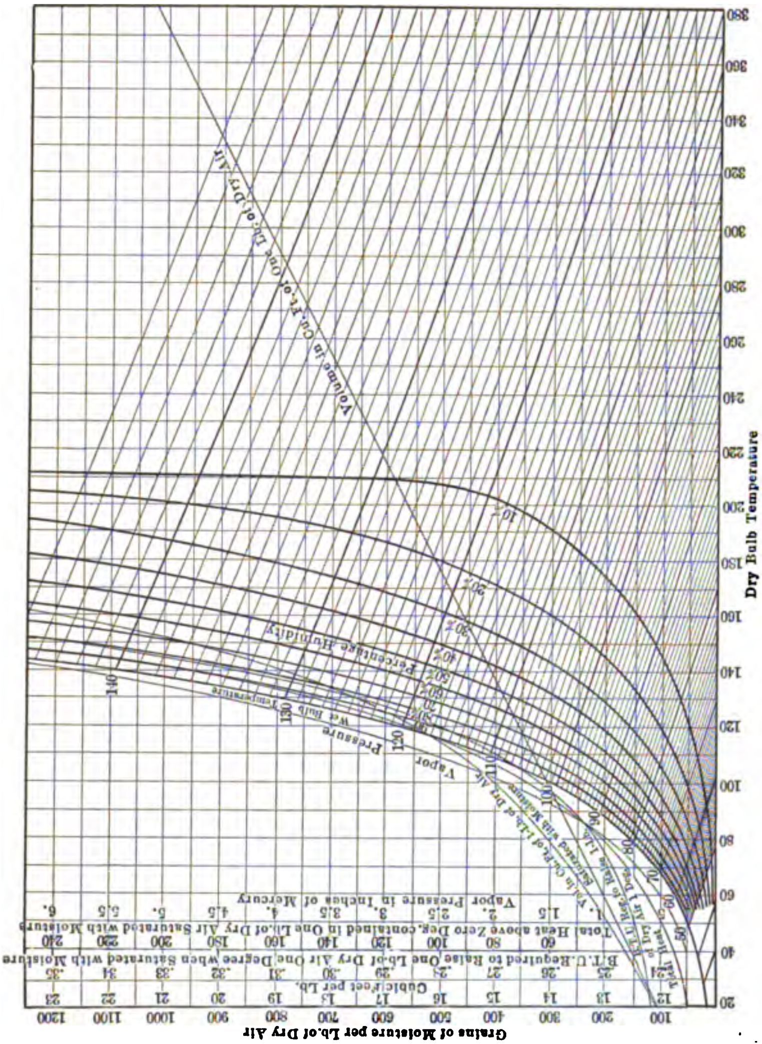

FIG. II.—Psychrometric chart.

STATIC PRESSURE TABLES FOR A. B. C. TYPE S, STEEL PLATE FAN CAPACITY TABLE

Table III.—No. 50 Single Inlet Steel Plate Fan—Type S

Volume	Outlet vel.	S. P. ¼"			S. P. ⅜"			S. P. ½"			S. P. ⅝"			S. P. ¾"			S. P. ⅞"		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
2250	1000	2366	301	.218	2690	343	.291	2940	375	.365	3175	404	.443	3400	433	.523	3610	460	.610
2475	1100	2490	317	.256	2780	354	.337	3040	387	.417	3267	416	.501	3480	443	.585	3670	468	.675
2700	1200	2600	331	.305	2925	373	.390	3125	398	.473	3360	427	.563	3575	455	.655	3763	480	.747
2925	1300	2736	348	.360	3000	382	.446	3237	412	.537	3475	442	.633	3675	468	.730	3865	492	.828
3150	1400	2846	362	.418	3107	395	.509	3310	422	.603	3573	455	.707	3750	478	.808	3965	505	.914
3375	1500	2987	381	.490	3226	411	.586	3460	440	.686	3650	465	.790	3860	492	.895	4060	517	1.01
3600	1600	3130	399	.563	3350	427	.666	3565	454	.770	3765	480	.880	3960	504	.990	4160	530	1.11
3825	1700	3270	416	.645	3475	442	.755	3680	469	.864	3885	495	.978	4055	517	1.10	4250	541	1.22
4050	1800	3410	434	.750	3607	460	.855	3810	485	.973	4010	510	1.09	4180	533	1.24	4350	554	1.34
4275	1900	3546	452	.838	3730	475	.965	3935	501	1.08	4120	524	1.21	4320	550	1.34	4455	567	1.46
4500	2000	3700	471	.949	3860	491	1.08	4050	515	1.20	4255	541	1.34	4423	564	1.47	4580	584	1.61
4725	2100	3850	490	1.07	4000	510	1.21	4210	536	1.33	4350	554	1.48	4535	578	1.62	4680	596	1.75
4950	2200	4000	510	1.20	4168	530	1.36	4320	550	1.49	4500	573	1.63	4670	595	1.78	4800	611	1.92
5175	2300				4323	550	1.50	4450	566	1.63	4628	589	1.79	4770	607	1.95	4930	628	2.10
5400	2400				4460	568	1.62	4620	588	1.82	4740	604	1.97	4920	626	2.13	5045	641	2.285
5625	2500				4600	586	1.83	4720	600	1.99	4880	622	2.16	5036	641	2.32	5170	658	2.485
5850	2600							4910	625	2.20	5000	637	2.36	5180	660	2.54	5325	678	2.700
6300	2800							5180	660	2.62	5280	671	2.81	5435	692	2.98	5510	702	3.180
6750	3000							5485	698	3.09	5610	714	3.31	5650	720	3.48	5840	744	3.692

Volume	Outlet vel.	S. P. 1"			S. P. 1¼"			S. P. 1½"			S. P. 1¾"			S. P. 2"			S. P. 2½"		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
2700	1200	3955	503	.850	4152	529	1.04	4470	570	1.26	4950	630	1.48	5230	667	1.72	5750	732	2.22
2925	1300	4050	515	.927	4380	558	1.13	4550	580	1.35	5024	640	1.59	5295	673	1.83	5820	740	2.34
3150	1400	4143	527	1.02	4465	569	1.24	4700	598	1.46	5105	650	1.70	5350	681	1.97	5900	752	2.49
3375	1500	4250	541	1.12	4570	582	1.33	4850	617	1.58	5180	660	1.83	5450	694	2.09	5950	757	2.64
3600	1600	4325	550	1.22	4652	594	1.47	4950	630	1.71	5245	667	1.96	5550	707	2.23	6025	767	2.80
3825	1700	4437	564	1.34	4750	605	1.61	5040	642	1.85	5330	679	2.11	5625	717	2.38	6100	777	2.96
4050	1800	4527	576	1.47	4846	616	1.73	5110	652	1.99	5410	689	2.28	5700	725	2.55	6195	788	3.15
4275	1900	4613	588	1.60	4945	630	1.87	5230	666	2.14	5520	702	2.44	5780	737	2.73	6265	798	3.33
4500	2000	4743	604	1.75	5075	646	2.03	5325	678	2.32	5620	715	2.62	5860	746	2.92	6365	810	3.53
4725	2100	4850	618	1.89	5145	655	2.19	5440	693	2.49	5724	729	2.81	5955	759	3.12	6475	825	3.76
4950	2200	4970	633	2.07	5256	670	2.365	5550	707	2.68	5790	738	3.00	6050	769	3.32	6550	835	4.00
5175	2300	5090	648	2.345	5370	684	2.565	5630	717	2.88	5900	751	3.22	6150	783	3.54	6610	844	4.24
5400	2400	5210	663	2.447	5480	698	2.770	5750	732	3.09	6025	767	3.44	6270	798	3.78	6700	853	4.50
5625	2500	5340	680	2.655	5610	715	2.970	5850	745	3.32	6100	776	3.67	6343	808	4.04	6800	865	4.76
5850	2600	5485	698	2.875	5740	731	3.22	5980	762	3.56	6200	790	3.92	6460	823	4.31	6880	877	5.04
6300	2800	5710	727	3.390	5960	759	3.715	6230	794	4.09	6460	822	4.48	6650	847	4.86	7090	903	5.65
6750	3000	5970	760	3.890	6200	790	4.28	6460	822	4.68	6675	850	5.07	6900	879	5.49	7295	928	6.32
7200	3200	6230	794	4.480	6475	825	4.91	6730	857	5.34	6920	881	5.74	7135	909	6.17	7530	960	7.02
7650	3400	6580	838	5.180	6740	858	5.62	6960	886	6.06	7150	910	6.51	7355	937	6.93	7750	987	7.83
8100	3600	6815	870	5.900	7000	894	6.37	7200	916	6.85	7440	948	7.30	7600	968	7.78	8020	1021	8.69
8550	3800	7105	905	6.730	7350	936	7.25	7475	952	7.72	7660	976	8.21	7840	999	8.67	8220	1047	9.68

CAPACITY TABLE

Table IV.—No. 60 Single Inlet Steel Plate Fan—Type S

Volume	Outlet vel.	S. P. ¼"			S. P. ⅜"			S. P. ½"			S. P. ⅝"			S. P. ¾"			S. P. ⅞"		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
3200	1000	2366	251	.287	2690	286	.415	2940	312	.519	3175	337	.630	3400	361	.743	3610	383	.868
3520	1100	2490	264	.365	2780	295	.478	3040	323	.593	3267	347	.712	3480	370	.832	3670	390	.960
3840	1200	2600	276	.435	2925	311	.554	3125	332	.673	3360	356	.800	3575	379	.932	3763	400	1.06
4160	1300	2736	290	.512	3000	319	.635	3237	344	.762	3475	369	.900	3675	390	1.04	3865	410	1.18
4480	1400	2846	302	.595	3107	330	.730	3310	351	.858	3573	379	1.00	3750	398	1.15	3965	421	1.30
4800	1500	2987	317	.697	3226	343	.833	3460	367	.977	3650	388	1.12	3860	410	1.27	4060	431	1.43
5120	1600	3130	332	.800	3350	356	.948	3565	379	1.09	3765	400	1.25	3960	420	1.41	4160	441	1.57
5440	1700	3270	347	.917	3475	369	1.07	3680	391	1.23	3885	413	1.39	4055	431	1.56	4250	451	1.73
5760	1800	3410	362	1.07	3607	382	1.21	3810	405	1.38	4010	425	1.55	4180	443	1.76	4350	462	1.92
6080	1900	3546	377	1.19	3730	396	1.37	3935	418	1.54	4120	437	1.72	4320	458	1.90	4455	473	2.08
6400	2000	3700	393	1.35	3860	410	1.53	4050	430	1.71	4255	452	1.91	4423	470	2.09	4580	487	2.29
6720	2100	3850	408	1.52	4000	425	1.72	4210	447	1.89	4350	462	2.11	4535	481	2.31	4680	498	2.49
7040	2200	4000	425	1.70	4168	443	1.93	4320	458	2.12	4500	478	2.32	4670	497	2.53	4800	510	2.73
7360	2300				4323	459	2.13	4450	473	2.33	4628	491	2.55	4770	507	2.77	4930	523	2.99
7680	2400				4460	473	2.31	4620	490	2.59	4740	504	2.80	4920	522	3.03	5045	537	3.25
8000	2500				4600	488	2.60	4720	502	2.83	4880	518	3.07	5036	534	3.30	5170	549	3.53
8320	2600							4910	521	3.13	5000	531	3.36	5180	550	3.61	5325	565	3.84
8960	2800							5180	550	3.71	5280	560	3.99	5435	578	4.23	5510	585	4.53
9600	3000							5485	582	4.40	5610	596	4.71	5650	600	4.96	5840	620	5.25

Volume	Outlet vel.	S. P. 1"			S. P. 1¼"			S. P. 1½"			S. P. 1¾"			S. P. 2"			S. P. 2½"		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
3840	1200	3955	420	1.21	4152	452	1.48	4470	475	1.79	4950	525	2.10	5230	555	2.44	5750	610	3.16
4160	1300	4050	430	1.32	4380	465	1.61	4550	483	1.92	5024	533	2.26	5295	561	2.61	5820	617	3.33
4480	1400	4143	439	1.45	4465	474	1.76	4700	499	2.08	5105	542	2.43	5350	568	2.80	5900	626	3.54
4800	1500	4250	451	1.59	4570	485	1.91	4850	515	2.25	5180	550	2.61	5450	578	2.97	5950	631	3.76
5120	1600	4325	459	1.74	4652	495	2.09	4950	526	2.43	5245	557	2.78	5550	589	3.18	6025	640	3.98
5440	1700	4437	471	1.91	4750	504	2.29	5040	534	2.63	5330	566	3.01	5625	598	3.39	6100	648	4.22
5760	1800	4527	481	2.08	4846	514	2.46	5110	542	2.83	5410	574	3.24	5700	605	3.63	6195	658	4.48
6080	1900	4613	490	2.27	4945	525	2.66	5230	555	3.05	5520	586	3.47	5780	613	3.89	6265	665	4.74
6400	2000	4743	504	2.48	5075	538	2.89	5325	565	3.29	5620	597	3.73	5860	621	4.16	6365	676	5.03
6720	2100	4850	515	2.69	5145	545	3.12	5440	578	3.55	5724	607	4.00	5955	632	4.43	6475	687	5.35
7040	2200	4970	528	2.94	5256	558	3.37	5550	589	3.81	5790	615	4.28	6050	642	4.72	6550	695	5.68
7360	2300	5090	540	3.33	5370	570	3.65	5630	598	4.09	5900	626	4.58	6150	653	5.05	6610	701	6.03
7680	2400	5210	553	3.48	5480	583	3.93	5750	610	4.39	6025	640	4.90	6270	666	5.38	6700	711	6.40
8000	2500	5340	567	3.78	5610	595	4.23	5850	621	4.72	6100	649	5.22	6343	674	5.74	6800	722	6.77
8320	2600	5485	582	4.09	5740	609	4.58	5980	635	5.07	6200	658	5.57	6460	686	6.13	6880	730	7.17
8960	2800	5710	606	4.82	5960	632	5.28	6230	661	5.83	6460	686	6.37	6650	706	6.91	7090	752	8.04
9600	3000	5970	633	5.54	6200	658	6.08	6460	686	6.67	6675	698	7.23	6900	732	7.82	7295	773	8.98
10240	3200	6230	662	6.37	6475	687	6.98	6730	715	7.58	6920	735	8.17	7135	757	8.80	7530	799	9.98
10880	3400	6580	698	7.36	6740	715	8.00	6960	739	8.62	7150	760	9.26	7355	781	9.87	7750	823	11.14
11520	3600	6815	723	8.40	7000	745	9.09	7200	764	9.75	7440	790	10.39	7600	807	11.08	8020	851	12.37
12160	3800	7105	755	9.58	7350	780	10.32	7475	793	10.98	7660	814	11.67	7840	832	12.33	8220	873	13.78

CAPACITY TABLE

TABLE V.—No. 70 SINGLE INLET STEEL PLATE FAN—TYPE S

Volume	Outlet vel.	S. P. ¼"			S. P. ⅜"			S. P. ½"			S. P. ⅝"			S. P. ¾"			S. P. ⅞"		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
4160	1000	2366	215	.402	2690	245	.538	2940	267	.674	3175	288	.818	3400	309	.965	3610	328	1.13
4576	1100	2490	228	.474	2780	253	.622	3040	276	.771	3267	297	.925	3480	316	1.08	3670	334	1.25
4992	1200	2600	236	.565	2925	266	.719	3125	284	.873	3360	305	1.04	3575	325	1.21	3763	342	1.38
5408	1300	2736	249	.665	3000	273	.825	3237	294	.992	3475	316	1.17	3675	334	1.35	3865	351	1.53
5824	1400	2846	258	.773	3107	283	.944	3310	301	1.11	3573	325	1.30	3750	341	1.49	3965	361	1.68
6240	1500	2987	271	.905	3226	293	1.08	3460	315	1.27	3650	332	1.46	3860	351	1.65	4060	370	1.86
6656	1600	3130	285	1.04	3350	305	1.23	3565	324	1.42	3765	341	1.62	3960	359	1.83	4160	379	2.04
7072	1700	3270	297	1.19	3475	316	1.39	3680	335	1.60	3885	353	1.81	4055	369	2.03	4250	386	2.25
7488	1800	3410	310	1.39	3607	328	1.58	3810	346	1.79	4010	365	2.01	4180	380	2.28	4350	396	2.49
7904	1900	3546	323	1.55	3730	339	1.78	3935	357	1.99	4120	375	2.23	4320	393	2.47	4455	405	2.70
8320	2000	3700	336	1.75	3860	351	1.99	4050	368	2.22	4255	387	2.48	4423	402	2.72	4580	417	2.97
8736	2100	3850	350	1.97	4000	364	2.23	4210	383	2.46	4350	396	2.73	4535	413	2.99	4680	426	3.24
9152	2200	4000	364	2.21	4168	379	2.50	4320	393	2.75	4500	410	3.02	4670	425	3.29	4800	436	3.55
9568	2300				4323	393	2.77	4450	405	3.02	4628	421	3.30	4770	433	3.60	4930	448	3.88
9984	2400				4460	406	2.99	4620	420	3.36	4740	430	3.63	4920	447	3.93	5045	459	4.22
10400	2500				4600	418	3.37	4720	430	3.68	4880	443	3.99	5036	458	4.28	5170	470	4.59
10816	2600							4910	446	4.07	5000	455	4.37	5180	471	4.69	5325	483	4.98
11648	2800							5180	471	4.82	5280	480	5.19	5435	491	5.50	5510	501	5.88
12480	3000							5485	499	5.71	5610	511	6.12	5650	514	6.43	5840	530	6.82

Volume	Outlet vel.	S. P. 1"			S. P. 1¼"			S. P. 1½"			S. P. 1¾"			S. P. 2"			S. P. 2½"		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
4992	1200	3955	359	1.57	4152	378	1.92	4470	407	2.33	4950	450	2.72	5230	475	3.17	5750	523	4.11
5408	1300	4050	368	1.71	4380	398	2.09	4550	415	2.50	5024	457	2.93	5295	481	3.39	5820	529	4.32
5824	1400	4143	376	1.88	4465	406	2.28	4700	427	2.71	5105	465	3.15	5350	487	3.63	5900	536	4.59
6240	1500	4250	386	2.07	4570	416	2.48	4850	441	2.91	5180	471	3.38	5450	495	3.86	5950	541	4.88
6656	1600	4325	393	2.26	4652	424	2.71	4950	450	3.15	5245	476	3.61	5550	505	4.13	6025	548	5.17
7072	1700	4437	404	2.47	4750	432	2.97	5040	459	3.42	5330	484	3.90	5620	511	4.41	6100	555	5.47
7488	1800	4527	412	2.70	4846	440	3.19	5110	465	3.67	5410	494	4.21	5700	519	4.72	6195	563	5.82
7904	1900	4613	420	2.95	4945	449	3.45	5230	475	3.96	5520	502	4.52	5780	525	5.05	6265	570	6.15
8320	2000	4743	430	3.22	5075	461	3.75	5325	484	4.28	5620	511	4.84	5860	533	5.40	6365	579	6.52
8736	2100	4850	441	3.50	5145	468	4.05	5440	494	4.61	5724	521	5.20	5955	541	5.76	6475	588	6.95
9152	2200	4970	452	3.81	5256	477	4.37	5550	505	4.96	5790	527	5.55	6050	550	6.13	6550	595	7.38
9568	2300	5090	463	4.33	5370	488	4.74	5630	512	5.32	5900	536	5.95	6150	560	6.55	6610	601	7.82
9984	2400	5210	474	4.52	5480	498	5.11	5750	523	5.70	6025	547	6.36	6270	570	6.98	6700	610	8.32
10400	2500	5340	485	4.91	5610	510	5.49	5850	532	6.13	6100	555	6.78	6343	575	7.46	6800	618	8.80
10816	2600	5485	498	5.32	5740	522	5.95	5980	544	6.58	6200	564	7.23	6460	587	7.96	6880	625	9.32
11648	2800	5710	520	6.26	5960	542	6.85	6230	567	7.57	6460	587	8.28	6650	605	8.98	7090	644	10.44
12480	3000	5970	542	7.19	6200	564	7.90	6460	598	8.66	6675	607	9.38	6900	627	10.13	7295	663	11.65
13312	3200	6230	567	8.27	6475	588	9.07	6730	612	9.86	6920	629	10.60	7135	648	11.40	7530	685	12.97
14144	3400	6580	598	9.58	6740	613	10.38	6960	632	11.19	7150	650	12.00	7355	670	12.81	7750	705	14.46
14976	3600	6815	620	10.88	7020	638	11.77	7200	655	12.66	7440	676	13.48	7600	691	14.38	8020	729	16.04
15808	3800	7105	646	12.43	7350	668	13.40	7475	680	14.25	7660	697	15.14	7840	713	16.00	8220	747	17.88

CAPACITY TABLE

TABLE VI.—No. 80 SINGLE INLET STEEL PLATE FAN—TYPE S

Volume	Outlet vel.	S. P. ¼"			S. P. ⅜"			S. P. ½"			S. P. ⅝"			S. P. ¾"			S. P. ⅞"		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
5050	1000	2366	189	.488	2690	214	.654	2940	234	.818	3175	253	.994	3400	271	1.17	3610	288	1.37
5555	1100	2490	198	.575	2780	222	.755	3040	242	.935	3267	260	1.12	3480	277	1.31	3670	292	1.52
6060	1200	2600	207	.685	2925	233	.873	3125	249	1.06	3360	268	1.26	3575	285	1.47	3763	300	1.68
6565	1300	2736	218	.808	3000	239	1.002	3237	257	1.21	3475	276	1.42	3675	292	1.63	3865	308	1.86
7070	1400	2846	227	.940	3107	248	1.144	3310	264	1.35	3573	285	1.58	3750	299	1.81	3965	316	2.05
7575	1500	2987	238	1.097	3226	257	1.314	3460	276	1.54	3650	291	1.77	3860	307	2.01	4060	324	2.26
8080	1600	3130	250	1.263	3350	267	1.497	3565	284	1.73	3765	298	1.97	3960	315	2.22	4160	332	2.48
8585	1700	3270	261	1.445	3475	277	1.695	3680	293	1.94	3885	310	2.19	4055	323	2.47	4250	339	2.74
9090	1800	3410	272	1.686	3607	287	1.920	3810	303	2.18	4010	319	2.44	4180	333	2.77	4350	347	3.02
9595	1900	3546	283	1.878	3730	297	2.165	3935	313	2.42	4120	328	2.71	4320	344	3.00	4455	350	3.28
10100	2000	3700	295	2.150	3860	305	2.425	4050	322	2.71	4255	339	3.01	4423	353	3.31	4580	365	3.61
10605	2100	3850	307	2.400	4000	319	2.71	4210	335	2.99	4350	347	3.33	4535	361	3.64	4680	373	3.94
11110	2200	4000	319	2.688	4168	332	3.04	4320	344	3.34	4500	358	3.67	4670	372	4.00	4800	383	4.32
11615	2300				4323	345	3.36	4450	354	3.67	4628	369	4.02	4770	380	4.37	4930	393	4.71
12120	2400				4460	356	3.63	4620	368	4.08	4740	378	4.42	4920	393	4.78	5045	402	5.12
12625	2500				4600	367	4.10	4720	376	4.47	4880	389	4.85	5036	401	5.20	5170	412	5.57
13130	2600							4910	392	4.94	5000	398	5.30	5180	413	5.69	5325	423	6.06
14140	2800							5180	413	5.85	5280	421	6.30	5435	433	6.67	5510	439	7.14
15150	3000							5485	437	6.93	5610	447	7.42	5650	450	7.81	5840	465	8.28

Volume	Outlet vel.	S. P. 1"			S. P. 1¼"			S. P. 1½"			S. P. 1¾"			S. P. 2"			S. P. 2½"		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m	B.hp.	Tip speed	R.p.m.	B.hp.
6060	1200	3955	315	1.91	4152	331	2.33	4470	356	2.83	4950	394	3.31	5230	417	3.86	5750	458	5.00
6565	1300	4050	322	2.08	4380	349	2.54	4550	363	3.03	5024	400	3.56	5295	421	4.12	5820	464	5.25
7070	1400	4143	329	2.28	4465	356	2.77	4700	375	3.29	5105	407	3.83	5350	426	4.42	5900	470	5.58
7575	1500	4250	338	2.51	4570	364	3.02	4850	386	3.54	5180	413	4.10	5450	434	4.68	5950	475	5.93
8080	1600	4326	345	2.75	4652	371	3.29	4950	395	3.83	5245	418	4.39	5550	442	5.02	6025	480	6.27
8585	1700	4437	353	3.01	4750	378	3.61	5040	402	4.15	5330	424	4.74	5625	448	5.35	6100	486	6.63
9090	1800	4527	361	3.29	4846	386	3.89	5110	407	4.46	5410	431	5.10	5700	455	5.73	6195	493	7.05
9595	1900	4613	368	3.59	4945	394	4.19	5230	417	4.81	5520	440	5.48	5780	460	6.13	6265	499	7.47
10100	2000	4743	377	3.91	5075	404	4.55	5325	424	5.19	5620	448	5.88	5860	467	6.57	6365	507	7.92
10605	2100	4850	386	4.25	5145	410	4.92	5440	433	5.60	5724	456	6.32	5955	475	7.00	6475	516	8.45
11110	2200	4970	396	4.64	5256	419	5.31	5550	443	6.02	5790	461	6.74	6050	482	7.45	6550	522	8.96
11615	2300	5090	405	5.36	5370	427	5.75	5630	448	6.45	5900	470	7.22	6150	490	7.95	6610	527	9.52
12120	2400	5210	416	5.48	5480	437	6.20	5750	458	6.92	6025	480	7.72	6270	500	8.48	6700	534	10.10
12625	2500	5340	425	5.96	5610	447	6.66	5850	466	7.45	6100	486	8.23	6343	505	9.06	6800	542	10.68
13130	2600	5485	437	6.44	5740	457	7.22	5980	477	7.98	6200	494	8.78	6460	515	9.67	6880	548	11.30
14140	2800	5710	455	7.60	5960	475	8.33	6230	497	9.19	6460	515	10.05	6650	530	10.90	7090	564	12.68
15150	3000	5970	476	8.73	6200	494	9.60	6460	515	10.53	6675	532	11.38	6900	550	12.33	7295	581	14.16
16160	3200	6230	497	10.05	6475	517	11.00	6730	537	11.97	6920	551	12.88	7135	568	13.86	7530	600	15.73
17179	3400	6580	524	11.62	6740	537	12.62	6960	555	13.60	7150	570	14.60	7355	587	15.55	7750	618	17.58
18180	3600	6815	543	13.23	7000	559	14.30	7200	574	15.40	7440	593	16.38	7600	605	17.48	8020	639	19.50
19190	3800	7105	565	15.10	7350	586	16.27	7475	596	17.30	7660	611	18.38	7840	624	19.44	8220	655	21.73

CAPACITY TABLE

Table VII.—No. 90 Single Inlet Steel Plate Fan—Type S

Volume	Outlet vel.	S. P. ¼″			S. P. ⅜″			S. P. ½″			S. P. ⅝″			S. P. ¾″			S. P. ⅞″		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
6450	1000	2366	167	.625	2690	190	.835	2940	208	1.05	3175	224	1.27	3400	240	1.49	3610	255	1.75
7095	1100	2490	176	.735	2780	196	.965	3040	215	1.19	3267	231	1.43	3480	246	1.68	3670	259	1.93
7740	1200	2600	184	.876	2925	207	1.11	3125	221	1.35	3360	238	1.61	3575	253	1.88	3763	266	2.14
8385	1300	2736	193	1.03	3000	212	1.27	3237	229	1.54	3475	245	1.81	3675	259	2.09	3865	273	2.37
9030	1400	2846	201	1.20	3107	220	1.46	3310	234	1.73	3573	253	2.03	3750	265	2.31	3965	280	2.61
9675	1500	2987	211	1.40	3226	228	1.68	3460	244	1.97	3650	258	2.26	3860	273	2.57	4060	287	2.89
10320	1600	3130	221	1.61	3350	237	1.91	3565	252	2.21	3765	266	2.52	3960	280	2.84	4160	294	3.17
10965	1700	3270	231	1.85	3475	245	2.17	3680	260	2.48	3885	275	2.81	4055	287	3.15	4250	300	3.49
11610	1800	3410	241	2.15	3607	255	2.45	3810	269	2.79	4010	283	3.12	4180	296	3.54	4350	308	3.86
12255	1900	3546	251	2.40	3730	264	2.76	3935	278	3.10	4120	291	3.46	4320	305	3.83	4455	315	4.19
12900	2000	3700	262	2.72	3860	273	3.09	4050	286	3.46	4255	301	3.86	4423	313	4.22	4580	324	4.62
13545	2100	3850	272	3.06	4000	283	3.47	4210	298	3.82	4350	308	4.25	4535	320	4.64	4680	331	5.03
14190	2200	1000	283	3.43	4168	295	3.88	4320	305	4.27	4500	318	4.68	1670	330	5.11	4800	340	5.52
14835	2300				4323	306	4.30	4450	314	4.69	4628	327	5.13	4770	338	5.59	4930	348	6.02
15480	2400				4460	315	4.64	4620	327	5.21	4740	335	5.65	1920	348	6.22	5045	356	6.55
16125	2500				4600	325	5.24	4720	334	5.71	4880	347	6.19	5036	356	6.64	5170	366	7.13
16770	2600							4910	348	6.31	5000	354	6.77	5180	366	7.28	5325	376	7.74
18060	2800							5180	366	7.47	5280	373	8.05	5435	384	8.53	5510	390	9.12
19350	3000							5485	387	8.87	5610	397	9.48	5650	400	9.98	5840	414	10.57

Volume	Outlet vel.	S. P. 1″			S. P. 1¼″			S. P. 1½″			S. P. 1¾″			S. P. 2″			S. P. 2½″		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
7740	1200	3955	279	2.44	4152	294	2.98	4470	316	3.61	4950	350	4.23	5230	370	4.93	5750	406	6.38
8385	1300	4050	286	2.65	4380	309	3.25	4550	322	3.88	5024	353	4.56	5295	374	5.26	5820	411	6.71
9030	1400	4143	293	2.92	4465	316	3.55	4700	332	4.21	5105	361	4.88	5350	378	5.65	5900	417	7.13
9675	1500	4250	300	3.21	4570	323	3.86	4850	343	4.53	5180	366	5.24	5450	385	5.98	5950	420	7.57
10320	1600	4325	306	3.51	4652	329	4.21	4950	350	4.89	5245	370	5.61	5550	392	6.41	6025	427	8.02
10965	1700	4437	313	3.84	4750	336	4.61	5040	356	5.31	5330	377	6.06	5625	398	6.83	6100	431	8.48
11610	1800	4527	320	4.20	4846	342	4.96	5110	362	5.70	5410	383	6.52	5700	403	7.32	6195	438	9.02
12255	1900	4613	327	4.58	4945	350	5.35	5230	370	6.15	5520	393	7.00	5780	408	7.82	6265	443	9.55
12900	2000	4743	335	5.00	5075	359	5.81	5325	377	6.63	5620	398	7.52	5860	415	8.38	6365	450	10.12
13545	2100	4850	343	5.43	5145	364	6.29	5440	384	7.15	5724	405	8.07	5955	421	8.94	6476	458	10.78
14190	2200	4970	352	5.82	5256	372	6.78	5550	393	7.70	5790	409	8.62	6050	428	9.52	6550	463	11.45
14835	2300	5090	360	6.61	5370	380	7.35	5630	398	8.25	5900	417	9.23	6150	435	10.15	6610	467	12.13
15480	2400	5210	369	7.01	5480	387	7.92	5750	406	8.85	6025	427	9.87	6270	442	10.84	6700	474	12.88
16125	2500	5340	377	7.62	5610	396	8.52	5850	413	9.52	6100	432	10.50	6343	449	11.96	6800	480	13.63
16770	2600	5485	388	8.25	5740	405	9.22	5980	424	10.20	6200	438	11.22	6460	456	12.35	6880	487	14.44
18060	2800	5710	404	9.72	5960	422	10.65	6230	441	11.74	6460	457	12.83	6650	470	13.73	7090	501	16.20
19350	3000	5970	422	11.15	6200	438	12.25	6460	457	13.45	6675	472	14.55	6900	488	15.73	7295	515	18.05
20640	3200	6230	441	12.83	6475	451	14.05	6730	476	15.30	6920	489	16.44	7135	503	17.70	7530	533	20.10
21930	3400	6580	465	14.85	6740	477	16.10	6960	492	17.35	7150	505	18.65	7355	519	19.86	7750	548	22.45
23220	3600	6815	482	16.90	7020	497	18.25	7200	510	19.65	7440	525	20.92	7600	538	22.30	8020	567	24.87
24510	3800	7105	503	19.30	7350	520	20.75	7475	528	22.13	7660	542	23.53	7840	554	24.85	8220	581	27.75

CAPACITY TABLE

Table VIII.—No. 100 Single Inlet Steel Plate Fan—Type S

Volume	Outlet vel.	S. P. ¼″			S. P. ⅜″			S. P. ½″			S. P. ⅝″			S. P. ¾″			S. P. ⅞″		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.ph.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
8260	1000	2306	150	.800	2690	171	1.07	2940	187	1.34	3175	202	1.62	3400	216	1.92	3610	230	2.24
9086	1100	2490	158	.942	2780	177	1.23	3040	193	1.53	3267	208	1.84	3480	221	2.15	3670	234	2.47
9912	1200	2600	165	1.12	2925	186	1.43	3125	199	1.73	3360	214	2.06	3575	227	2.40	3763	240	2.74
10738	1300	2736	174	1.32	3000	191	1.64	3237	206	1.97	3475	221	2.37	3675	233	2.67	3865	246	3.03
11564	1400	2846	181	1.53	3107	198	1.87	3310	211	2.21	3573	227	2.59	3750	239	2.96	3965	252	3.35
12390	1500	2987	190	1.79	3226	205	2.14	3460	220	2.52	3650	233	2.90	3860	246	3.28	4060	258	3.69
13216	1600	3130	199	2.06	3350	213	2.44	3565	227	2.82	3765	240	3.23	3960	252	3.63	4160	265	4.07
14042	1700	3270	208	2.37	3475	222	2.77	3680	234	3.17	3885	247	3.59	4055	256	4.03	4250	270	4.47
14868	1800	3410	217	2.75	3607	230	3.14	3810	242	3.57	4010	255	3.99	4180	266	4.53	4350	277	4.95
15694	1900	3546	226	3.07	3730	237	3.54	3935	252	3.97	4120	262	4.43	4320	275	4.90	4455	284	5.37
16520	2000	3700	235	3.47	3860	245	3.97	4050	258	4.43	4255	269	4.93	4423	282	5.41	4580	292	5.90
17346	2100	3850	245	3.92	4000	254	4.43	4210	268	4.88	4350	277	5.44	4535	289	5.95	4680	298	6.44
18172	2200	4000	254	4.39	4168	265	4.97	4320	275	5.47	4500	286	6.00	4670	297	6.54	4800	306	7.06
18998	2300				4323	275	5.50	4450	283	6.01	4628	294	6.57	4770	303	7.15	4930	314	7.70
19824	2400				4460	284	5.95	4620	294	6.67	4740	302	7.23	4920	313	7.82	5045	321	8.38
20650	2500				4600	293	6.70	4720	301	7.32	4880	310	7.92	5036	320	8.52	5170	329	9.13
21476	2600							4910	312	9.08	5000	318	8.52	5180	329	9.33	5325	339	9.92
23128	2800							5180	330	9.57	5280	338	10.30	5435	346	10.92	5510	351	11.67
24780	3000							5485	349	11.34	5610	357	12.14	5650	359	12.77	5840	371	13.53

Volume	Outlet vel.	S. P. 1″			S. P. 1¼″			S. P. 1½″			S. P. 1¾″			S. P 2″			S. P. 2½″		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
9912	1200	3955	251	3.12	4152	264	3.82	4470	285	4.62	4950	315	5.42	5230	333	6.36	5750	366	8.17
10738	1300	4050	258	3.40	4380	278	4.16	4550	290	4.96	5024	320	5.83	5295	337	6.72	5820	370	8.60
11564	1400	4143	263	3.74	4465	285	4.54	4700	299	5.38	5105	325	6.25	5350	340	7.22	5900	375	9.13
12390	1500	4250	270	4.11	4570	291	4.93	4850	308	5.80	5180	329	6.72	5450	347	7.67	5950	379	9.72
13216	1600	4325	275	4.50	4652	297	5.38	4950	315	6.27	5245	334	7.18	5550	353	8.20	6025	383	10.26
14042	1700	4437	282	4.92	4750	302	5.90	5040	321	6.79	5330	339	7.75	5625	358	8.76	6100	388	10.85
14868	1800	4527	288	5.38	4846	308	6.36	5110	325	7.29	5410	344	8.36	5700	363	9.37	6195	394	11.55
15694	1900	4613	294	5.86	4945	314	6.85	5230	333	7.87	5520	351	8.97	5780	368	10.02	6265	398	12.22
16520	2000	4743	301	6.41	5075	323	7.44	5325	339	8.50	5620	357	9.63	5860	373	10.73	6365	405	12.97
17346	2100	4850	308	6.95	5145	328	8.05	5440	346	9.17	5724	364	10.32	5955	379	11.44	6475	412	13.80
18172	2200	4970	316	7.58	5256	334	8.68	5550	354	9.85	5790	368	11.03	6050	385	12.18	6550	417	14.65
18998	2300	5090	324	8.60	5370	342	9.42	5630	358	10.55	5900	375	11.82	6150	391	13.00	6610	421	15.54
19824	2400	5210	332	8.97	5480	349	10.15	5750	366	11.32	6025	383	12.64	6270	399	13.86	6700	426	16.51
20650	2500	5340	340	9.75	5610	357	10.90	5850	372	12.18	6100	388	13.46	6343	403	14.80	6800	432	17.48
21476	2600	5485	349	10.55	5740	366	11.80	5980	381	13.06	6200	394	14.37	6460	411	15.80	6880	438	18.50
23128	2800	5710	364	12.43	5960	379	13.62	6230	396	15.03	6460	411	16.42	6650	423	17.85	7090	451	20.73
24780	3000	5970	380	14.28	6200	395	15.68	6460	411	17.20	6675	425	18.63	6900	439	20.15	7295	464	23.15
26432	3200	6230	396	16.43	6475	412	18.00	6730	428	19.60	6920	441	21.07	7135	448	22.67	7530	478	25.73
28084	3400	6580	418	19.00	6740	429	20.60	6960	443	22.25	7150	455	23.88	7355	469	25.40	7750	493	28.70
29736	3600	6815	433	21.65	7020	447	23.35	7200	458	25.13	7440	473	26.75	7600	484	28.60	8020	511	31.90
31388	3800	7105	452	24.70	7350	468	26.60	7475	476	28.30	7660	488	30.10	7840	499	31.80	8220	523	35.55

CAPACITY TABLE

Table IX.—No. 110 Single Inlet Steel Plate Fan—Type S

Volume	Outlet vel.	S. P. ¼″			S. P. ⅜″			S. P. ½″			S. P. ⅝″			S. P. ¾″			S. P. ⅞″		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
9760	1000	2366	137	.945	2690	156	1.26	2940	170	1.58	3175	184	1.92	3400	197	2.26	3610	209	2.65
10736	1100	2490	144	1.11	2780	161	1.46	3040	176	1.81	3267	190	2.17	3480	202	2.53	3670	212	2.93
11712	1200	2600	151	1.32	2925	169	1.69	3125	181	2.05	3360	195	2.44	3575	207	2.84	3763	218	3.24
12688	1300	2736	157	1.56	3000	174	1.93	3237	187	2.32	3475	201	2.74	3675	213	3.16	3865	224	3.58
13664	1400	2846	165	1.81	3107	180	2.21	3310	192	2.61	3573	207	3.06	3750	217	3.50	3965	230	3.96
14640	1500	2987	173	2.12	3226	187	2.54	3460	200	2.97	3650	211	3.43	3860	224	3.88	4060	235	4.37
15616	1600	3130	183	2.44	3350	194	2.89	3565	207	3.34	3765	218	3.81	3960	229	4.29	4160	241	4.80
16592	1700	3270	189	2.79	3475	201	3.27	3680	213	3.74	3885	225	4.24	4055	235	4.76	4250	246	5.28
17568	1800	3410	197	3.25	3607	209	3.71	3810	221	4.22	4010	232	4.72	4180	242	5.36	4350	252	5.84
18544	1900	3546	206	3.63	3730	216	4.18	3935	228	4.68	4120	239	5.23	4320	250	5.79	4455	258	6.34
19520	2000	3700	214	4.11	3860	224	4.68	4050	235	5.22	4255	245	5.82	4423	257	6.39	4580	265	6.97
20496	2100	3850	223	4.63	4000	232	5.24	4210	244	5.77	4350	252	6.42	4535	262	7.02	4680	271	7.61
21472	2200	4000	232	5.18	4168	242	5.87	4320	251	6.46	4500	261	7.08	4670	271	7.72	4800	278	8.34
22448	2300				4323	251	6.50	4450	258	7.10	4628	268	7.76	4770	277	8.45	4930	286	9.12
23424	2400				4460	258	7.02	4620	268	7.88	4740	275	8.55	4920	285	9.24	5045	292	9.92
24400	2500				4600	266	7.93	4720	273	8.63	4880	283	9.37	5036	292	10.05	5170	300	10.77
25376	2600							4910	284	9.55	5000	290	10.25	5180	300	11.00	5325	308	11.72
27328	2800							5180	300	11.3	5280	306	12.17	5435	315	12.88	5510	319	13.80
29280	3000							5485	317	13.4	5610	325	14.34	5650	327	15.08	5840	338	16.00

Volume	Outlet vel.	S. P. 1″			S. P. 1¼″			S. P. 1½″			S. P. 1¾″			S. P. 2″			S. P. 2½″		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
11712	1200	3955	229	3.69	4152	241	4.52	4470	259	5.47	4950	287	6.40	5230	303	7.45	5750	333	9.64
12688	1300	4050	235	4.02	4380	254	4.92	4550	264	5.87	5024	291	6.88	5295	306	7.95	5820	337	10.15
13664	1400	4143	240	4.42	4465	259	5.37	4700	272	6.36	5105	296	7.38	5350	310	8.53	5900	342	10.78
14640	1500	4250	246	4.85	4570	265	5.83	4850	281	6.85	5180	300	7.93	5450	316	9.06	5950	345	11.46
15616	1600	4325	250	5.32	4652	270	6.37	4950	287	7.40	5245	303	8.50	5550	322	9.69	6025	349	12.13
16592	1700	4437	257	5.81	4750	275	6.97	5040	292	8.02	5330	309	9.17	5625	326	10.34	6100	353	12.83
17568	1800	4527	262	6.35	4846	280	7.50	5110	296	8.62	5410	313	9.86	5700	330	11.07	6195	358	13.65
18544	1900	4613	267	6.92	4945	286	8.10	5230	303	9.30	5520	320	10.58	5780	335	11.84	6265	363	14.44
19520	2000	4743	274	7.57	5075	294	8.80	5325	309	10.03	5620	325	11.37	5860	340	12.68	6365	369	15.32
20496	2100	4850	281	8.22	5145	298	9.52	5440	315	10.83	5724	332	12.20	5955	345	13.52	6475	375	16.30
21472	2200	4970	288	8.96	5256	305	10.24	5550	322	11.63	5790	335	13.02	6050	350	14.40	6550	379	17.30
22448	2300	5090	295	10.15	5370	311	11.11	5630	326	12.48	5900	342	13.96	6150	356	15.37	6610	383	18.37
23424	2400	5210	302	10.62	5480	312	11.99	5750	333	13.39	6025	349	14.93	6270	363	16.39	6700	388	19.54
24400	2500	5340	309	11.52	5610	325	12.88	5850	339	14.38	6100	353	15.90	6343	367	17.50	6800	394	20.65
25376	2600	5485	318	12.47	5740	332	13.96	5980	346	15.43	6200	359	16.97	6460	375	18.70	6880	399	21.86
27328	2800	5710	331	14.68	5960	345	16.10	6230	361	17.75	6460	375	19.43	6650	385	21.08	7090	405	24.50
29280	3000	5970	346	16.87	6200	359	18.55	6460	374	20.39	6675	387	22.00	6900	400	23.80	7295	423	27.33
31232	3200	6230	361	19.43	6475	375	21.30	6730	390	23.10	6920	401	24.90	7135	413	26.75	7530	437	30.46
33184	3400	6580	381	22.45	6740	390	24.35	6960	403	26.55	7150	414	28.20	7355	427	30.10	7750	449	33.95
35136	3600	6815	395	25.55	7020	407	27.60	7200	417	29.70	7440	431	31.60	7600	440	33.75	8020	465	37.70
37088	3800	7105	412	29.15	7350	426	31.47	7475	433	33.35	7660	444	35.55	7840	454	37.55	8220	476	42.00

CAPACITY TABLE

TABLE X.—No. 120 SINGLE INLET STEEL PLATE FAN—TYPE S

Volume	Outlet vel.	S. P. ¼″			S. P. ⅜″			S. P. ½″			S. P. ⅝″			S. P. ¾″			S. P. ⅞″		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
11950	1000	2366	125	1.156	2690	143	1.54	2940	156	1.44	3175	168	2.35	3400	180	2.77	3610	191	3.24
13145	1100	2490	132	1.36	2780	147	1.79	3040	161	2.21	3267	173	2.66	3480	185	3.11	3670	195	3.59
14340	1200	2600	138	1.60	2925	155	2.07	3125	166	2.51	3360	178	2.99	3575	189	3.48	3763	200	3.97
15535	1300	2736	145	1.91	3000	159	2.36	3237	172	2.85	3475	184	3.36	3675	195	3.87	3865	205	4.39
16730	1400	2846	151	2.23	3107	165	2.71	3310	176	3.20	3573	189	3.75	3750	199	4.29	3965	210	4.85
17925	1500	2987	158	2.60	3226	171	3.11	3460	184	3.64	3650	194	4.20	3860	205	4.76	4060	215	5.35
19120	1600	3130	166	2.99	3350	178	3.54	3565	189	4.08	3765	200	4.67	3960	210	5.26	4160	220	5.87
20315	1700	3270	173	3.42	3475	184	4.02	3680	195	4.61	3885	206	5.19	4055	215	5.88	4250	226	6.48
21510	1800	3410	181	3.99	3607	191	4.54	3810	202	5.17	4010	213	5.79	4180	222	6.56	4350	231	7.16
22705	1900	3546	188	4.45	3730	198	5.12	3935	209	5.74	4120	219	6.42	4320	229	7.10	4455	236	7.77
23900	2000	3700	196	5.04	3860	205	5.74	4050	215	6.40	4255	226	7.14	4423	235	7.83	4580	243	8.54
25095	2100	3850	204	5.68	4000	212	6.42	4210	223	7.08	4350	231	7.87	4535	241	8.61	4680	248	9.33
26290	2200	4000	212	6.36	4168	221	7.19	4320	229	7.92	4500	239	8.68	4670	248	9.47	4800	254	10.22
27485	2300				4323	230	8.08	4450	236	8.70	4628	245	9.51	4770	253	10.35	4930	262	11.15
28680	2400				4460	237	8.62	4620	245	9.65	4740	251	10.45	4920	261	11.33	5045	268	12.14
39375	2500				4600	244	9.70	4720	251	10.58	4880	259	11.48	5036	267	12.32	5170	275	13.20
31070	2600							4910	261	11.70	5000	265	12.55	5180	275	13.48	5325	282	14.35
33460	2800							5180	275	13.87	5280	280	14.92	5435	288	15.80	5510	292	16.88
35850	3000							5485	291	16.40	5610	298	17.88	5650	300	18.48	5840	310	19.60

Volume	Outlet vel.	S. P. 1″			S. P. 1¼″			S. P. 1½″			S. P. 1¾″			S. P. 2″			S. P. 2½″		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
14340	1200	3955	209	4.52	4152	220	5.52	4470	237	6.69	4950	262	7.83	5230	279	9.12	5750	305	11.73
15535	1300	4050	215	4.92	4380	232	6.02	4550	242	7.18	5024	267	8.43	5295	281	9.75	5820	309	12.4
16730	1400	4143	220	5.40	4465	237	6.57	4700	249	7.80	5105	271	9.04	5350	284	10.44	5900	313	13.2
17925	1500	4250	226	5.95	4570	243	7.15	4850	257	8.40	5180	275	9.73	5450	289	11.10	5950	316	14.0
19120	1600	3325	229	6.50	4652	247	7.80	4950	263	9.08	5245	278	10.40	5550	294	11.87	6025	320	14.8
20315	1700	4437	235	7.12	4750	252	8.55	5040	268	9.84	5330	283	11.22	5625	298	12.1	6100	324	15.7
21510	1800	4527	240	7.78	4846	257	9.20	5110	271	10.56	5410	287	12.10	5700	302	13.6	6195	328	16.7
22705	1900	4613	245	8.48	4945	262	9.92	5230	277	11.33	5520	293	12.9	5780	307	14.5	6266	333	17.7
23900	2000	4743	251	9.27	5075	269	10.77	5325	283	12.3	5620	298	13.9	5860	311	15.5	6365	338	18.7
25095	2100	4850	257	10.07	5145	273	11.67	5440	289	13.2	5724	304	14.9	5955	316	16.5	6475	344	20.0
26290	2200	4970	264	10.97	5256	279	12.56	5550	294	14.2	5790	307	15.9	6050	321	17.6	6550	348	21.2
27485	2300	5090	270	12.45	5370	285	13.60	5630	299	15.3	5900	313	17.1	6150	326	18.8	6610	351	22.5
28680	2400	5210	276	12.98	5480	292	14.70	5750	305	16.4	6025	320	18.3	6270	333	20.1	6700	356	23.9
29875	2500	5340	283	14.12	5610	298	15.78	5850	310	17.6	6100	324	19.5	6343	336	21.5	6800	361	25.3
31070	2600	5485	291	15.27	5740	304	17.10	5980	317	18.9	6200	329	20.8	6460	343	22.9	6880	365	26.7
33460	2800	5710	303	18.00	5960	316	19.73	6230	331	21.7	6460	343	23.8	6650	353	25.8	7090	376	30.0
35850	3000	5970	317	20.68	6200	329	22.70	6460	343	24.9	6675	354	26.9	6900	366	29.2	7295	387	33.5
38240	3200	6230	330	23.80	6475	344	26.10	6730	357	28.3	6920	367	30.5	7135	378	32.8	7530	399	37.2
40630	3400	6580	349	27.50	6740	357	29.85	6960	320	32.1	7150	379	34.6	7355	391	36.8	7750	411	41.7
43020	3600	6815	362	31.30	7000	372	33.80	7200	382	36.4	7440	394	38.7	7600	404	41.4	8020	426	46.2
45410	3800	7105	377	35.80	7350	390	38.50	7475	397	41.0	7660	407	43.6	7840	417	46.0	8220	436	51.4

CAPACITY TABLE

Table XI.—No. 130 Single Inlet Steel Plate Fan—Type S

Volume	Outlet vel.	S. P. ¼"			S. P. ⅜"			S. P. ½"			S. P. ⅝"			S. P. ¾"			S. P. ⅞"		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
14050	1000	2366	116	1.360	2690	132	1.820	2940	144	2.280	3175	156	2.765	3400	166	3.262	3610	177	3.810
15455	1100	2490	122	1.602	2780	136	2.101	3040	149	2.607	3267	160	3.128	3480	171	3.658	3670	180	4.218
16860	1200	2600	127	1.909	2925	143	2.433	3125	153	2.952	3360	165	3.516	3575	175	4.091	3763	184	4.668
18265	1300	2736	134	2.250	3060	147	2.790	3237	157	3.351	3475	170	3.950	3675	180	4.555	3865	189	5.162
19670	1400	2846	139	2.620	3107	152	3.190	3310	162	3.771	3573	175	4.414	3750	184	5.050	3965	194	5.710
21075	1500	2987	146	3.060	3226	158	3.660	3460	170	4.290	3650	179	4.937	3860	189	5.595	4060	199	6.290
22480	1600	3130	154	3.515	3350	164	4.168	3565	175	4.807	3765	185	5.500	3960	193	6.190	4160	204	6.918
23885	1700	3270	160	4.027	3475	170	4.717	3680	180	5.408	3885	190	6.102	4055	197	6.868	4250	208	7.620
25290	1800	3410	167	4.690	3607	177	5.345	3810	187	6.078	4010	196	6.800	4180	205	7.715	4350	213	8.423
26695	1900	3546	174	5.230	3730	183	6.020	3935	193	6.752	4120	202	7.550	4320	212	8.350	4455	218	9.147
28100	2000	3700	181	5.935	3860	189	6.250	4050	198	7.540	4255	209	8.400	4423	217	9.210	4580	225	10.04
29505	2100	3850	189	6.678	4000	196	7.550	4210	206	8.320	4350	213	9.253	4535	222	10.120	4680	229	10.97
30910	2200	4000	196	7.475	4168	204	8.452	4320	212	9.300	4500	221	10.20	4670	229	11.120	4800	235	12.02
32315	2300				4323	212	9.230	4450	218	10.23	4628	227	11.18	4770	234	12.170	4930	242	13.12
33720	2400				4460	219	10.130	4620	226	11.34	4740	232	12.30	4920	241	13.30	5045	247	14.26
35125	2500				4600	226	11.40	4720	231	12.43	4880	239	13.48	5036	247	14.48	5170	253	15.49
36530	2600							4910	241	13.75	5000	245	14.76	5180	254	15.85	5325	261	16.87
39340	2800							5180	254	16.29	5280	259	17.53	5435	266	18.60	5410	270	19.87
42150	3000							5485	269	19.30	5610	275	20.66	5650	277	21.74	5840	286	23.06

Volume	Outlet vel.	S. P. 1"			S. P. 1¼"			S. P. 1½"			S. P. 1¾"			S. P. 2"			S. P. 2¼"		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
16860	1200	3955	194	5.303	4152	204	6.501	4470	219	7.875	4950	243	9.32	5230	256	10.73	5750	282	13.91
18265	1300	4050	198	5.790	4380	215	7.085	4550	223	8.351	5024	246	9.93	5295	259	11.46	5820	285	14.63
19670	1400	4143	203	6.355	4465	219	7.337	4700	231	9.160	5105	250	10.63	5350	262	12.28	5900	289	15.53
21075	1500	4250	208	7.000	4570	224	8.400	4850	238	9.870	5180	254	11.43	5450	267	13.05	5950	291	16.50
22480	1600	4325	212	7.650	4652	229	9.170	4950	243	10.67	5245	257	12.23	5550	272	13.45	6025	295	17.49
23855	1700	4437	217	8.375	4750	233	10.04	5040	247	11.56	5330	262	13.20	5625	276	14.90	6100	299	18.48
25290	1800	4527	222	9.150	4846	238	10.80	5110	251	12.41	5410	265	14.25	5700	279	15.94	6195	303	19.65
26695	1900	4613	226	9.980	4945	243	11.67	5230	256	13.38	5520	270	15.27	5780	283	17.05	6265	307	20.83
28100	2000	4743	232	10.90	5075	249	12.67	5325	261	14.46	5620	275	16.40	5860	287	19.27	6365	312	22.10
29505	2100	4850	238	11.83	5145	252	13.70	5440	267	15.59	5724	280	17.88	5955	292	19.50	6425	317	23.50
30910	2200	4970	243	12.90	5256	257	14.78	5550	272	16.77	5790	284	18.78	6050	296	20.73	6550	321	24.94
32315	2300	5090	249	14.63	5370	263	16.00	5630	276	17.99	5900	289	20.10	6150	301	22.13	6610	324	26.46
33720	2400	5210	255	15.27	5480	269	17.27	5750	282	19.29	6025	295	21.50	6270	307	23.60	6700	329	28.10
35125	2500	5340	262	16.59	5610	275	18.55	5850	287	20.73	6100	299	22.90	6343	311	25.23	6800	333	29.73
36530	2600	5485	269	17.96	5740	281	20.12	5980	293	22.24	6200	304	24.38	6460	316	26.90	6880	339	31.50
39340	2800	5710	280	21.14	5960	292	23.20	6230	305	25.60	6460	317	28.00	6650	326	30.35	7090	342	35.30
42150	3000	5970	292	24.32	6200	303	26.72	6460	317	29.30	6675	327	31.70	6900	338	34.30	7295	357	39.40
44960	3200	6230	305	28.00	6475	317	30.63	6730	330	33.32	6920	339	35.90	7135	349	38.60	7593	369	43.80
47770	3400	6580	322	32.35	6740	330	35.08	6960	341	37.80	7150	351	40.70	7355	361	43.30	7750	380	49.00
50580	3600	6815	335	36.82	7020	344	39.80	7200	353	42.80	7440	365	45.60	7600	373	48.70	8020	395	54.30
53390	3800	7105	349	42.02	7350	360	45.37	7475	367	48.20	7660	375	51.20	7840	384	54.20	8220	403	60.50

CAPACITY TABLE

Table XII.—No. 140 Single Inlet Steel Plate Fan—Type S

Vol-ume	Outlet vel.	S. P. ¼"			S. P. ⅜"			S. P. ½"			S. P. ⅝"			S. P. ¾"			S. P. ⅞"		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
16000	1000	2366	108	1.550	2690	123	2.072	2940	134	2.596	3175	145	3.150	3400	155	3.715	3610	164	4.337
17600	1100	2490	113	1.825	2780	127	2.392	3040	138	2.967	3267	149	3.560	3480	158	4.160	3670	167	4.800
19200	1200	2600	118	2.172	2925	133	2.770	3125	142	3.360	3360	153	4.000	3575	163	4.655	3763	171	5.318
20800	1300	2736	124	2.560	3000	137	3.175	3237	147	3.817	3475	158	4.500	3675	167	5.187	3865	176	58.80
22400	1400	2846	129	2.980	3107	141	3.630	3310	151	4.299	3573	163	5.025	3750	171	5.750	3965	180	65.10
24000	1500	2987	136	3.482	3226	147	4.168	3460	157	4.885	3650	166	5.620	3860	176	6.370	4060	185	7.160
25600	1600	3130	142	4.000	3350	153	4.747	3565	162	5.475	3765	171	6.255	3960	180	7.045	4160	189	7.870
27200	1700	3270	149	4.585	3475	158	5.368	3680	168	6.155	3885	177	6.950	4055	184	7.820	4250	193	86.70
28800	1800	3410	155	5.340	3607	164	6.087	3810	173	6.920	4010	183	7.750	4180	190	8.787	4350	198	9.590
30400	1900	3546	161	5.950	3730	170	6.850	3935	179	7.699	4120	187	8.600	4320	197	9.510	4455	203	10.4
32000	2000	3700	168	6.750	3860	176	7.690	4050	184	8.580	4255	194	9.560	4423	201	10.5	4580	209	11.4
33600	2100	3850	175	7.600	4000	182	8.600	4210	191	9.475	4350	198	10.5	4535	206	11.5	4680	213	12.5
35200	2200	4000	182	8.520	4168	189	9.625	4320	197	10.6	4500	205	11.6	4670	213	12.7	4800	219	13.7
36800	2300				4323	197	10.6	4450	203	11.6	4628	210	12.7	4770	217	13.9	4930	224	15.0
38400	2400				4460	203	11.5	4620	210	12.9	4740	216	14.0	4920	224	15.2	5045	229	16.24
40000	2500				4600	209	13.0	4720	215	14.2	4880	222	15.3	5036	229	16.49	5170	235	17.62
41600	2600							4910	223	15.6	5000	237	16.80	5180	236	18.05	5325	242	19.20
44800	2800							5180	236	18.55	5280	240	19.98	5435	247	21.18	5510	251	22.62
48000	3000							5485	249	21.96	5610	255	23.53	5650	257	24.76	5840	226	26.25

Vol-ume	Outlet vel.	S. P. 1"			S. P. 1¼"			S. P. 1½"			S. P. 1¾"			S. P. 2"			S. P. 2½"		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
19200	1200	3955	180	6.045	4152	189	7.400	4470	203	8.965	4950	225	10.5	5230	236	12.2	5750	262	15.8
20300	1300	4050	184	6.595	4380	199	8.070	4550	207	9.637	5024	228	11.3	5295	241	13.0	5820	265	16.64
22400	1400	4143	188	7.247	4465	203	8.808	4700	212	10.4	5105	232	12.1	5350	243	14.0	5900	263	17.68
24000	1500	4250	193	7.957	4570	208	9.560	4850	221	11.2	5180	236	13.0	5450	248	14.9	5950	271	18.80
25600	1600	4325	197	8.710	4652	212	10.4	4950	225	12.1	5245	239	13.9	5550	252	15.9	6025	274	19.90
27200	1700	4437	201	9.545	4750	216	11.4	5040	229	13.2	5330	243	15.0	5625	256	16.98	6100	277	21.00
28800	1800	4527	206	10.4	4846	221	12.3	5110	232	14.2	5410	246	16.20	5700	259	18.16	6195	282	22.35
30400	1900	4613	210	11.4	4945	225	13.3	5230	238	15.3	5520	251	17.37	5780	263	19.43	6265	285	23.68
32000	2000	4743	215	12.4	5075	231	14.4	5325	242	16.45	5620	255	18.65	5860	267	20.80	6365	289	25.10
33600	2100	4850	221	13.5	5145	234	15.6	5440	247	17.73	5724	260	20.00	5955	271	22.18	6475	294	26.80
35200	2200	4970	226	14.7	5256	239	16.82	5550	252	19.10	5790	264	21.38	6050	275	23.61	6550	298	28.40
36800	2300	5090	231	16.67	5370	244	18.21	5630	256	20.47	5900	268	22.90	6150	280	25.20	6610	301	30.20
38400	2400	5210	237	17.39	5480	249	19.67	5750	261	22.00	6025	274	24.46	6270	285	26.90	6700	305	32.00
40000	2500	5340	243	18.89	5610	255	21.12	5850	266	23.60	6100	277	26.10	6343	289	28.72	6800	309	33.80
41600	2600	5485	249	20.46	5743	261	22.90	5980	272	25.35	6200	282	27.85	6460	293	30.65	6880	313	35.80
44800	2800	5710	260	24.08	5960	271	26.41	6230	283	29.15	6460	293	31.85	6650	303	34.60	7090	322	40.10
48000	3000	5970	272	27.70	6200	282	30.40	6460	294	33.36	6675	304	36.10	6900	314	39.08	7295	332	44.90
51200	3200	6230	283	31.90	6475	294	34.90	6730	306	37.95	6920	315	40.80	7135	324	43.90	7530	343	50.00
54400	3400	6580	299	36.80	6740	307	39.92	6960	316	43.07	7150	325	46.25	7355	335	49.30	7750	353	55.70
57600	3600	6815	310	41.92	7020	319	45.30	7200	327	48.07	7440	339	51.99	7600	346	55.38	8020	365	61.80
60800	3800	7105	323	47.90	7350	334	51.60	7475	340	54.92	7660	349	58.37	7840	357	61.60	8220	374	69.00

CAPACITY TABLE

Table XIII.—No. 160 Single Inlet Steel Plate Fan—Type S

Volume	Outlet vel.	S. P. ¼″			S. P. ⅜″			S. P. ½″			S. P. ⅝″			S. P. ¾″			S. P. ⅞″		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
20250	1000	2366	94	1.957	2690	107	2.615	2940	117	3.28	3175	127	3.98	3400	135	4.69	3610	144	5.48
22275	1100	2490	99	2.31	2780	111	3.025	3040	121	3.75	3267	130	4.5	3480	139	5.25	3670	146	6.08
24300	1200	2600	104	2.75	2925	116	3.505	3125	125	4.25	3360	134	5.06	3575	142	5.89	3763	150	6.72
26325	1300	2736	109	3.23	3060	119	4.01	3237	129	4.82	3475	138	5.68	3675	146	6.55	3865	154	7.44
28350	1400	2846	113	3.77	3107	124	4.59	3310	132	5.43	3573	142	6.35	3750	149	7.26	3965	158	8.2
30375	1500	2987	119	4.40	3226	128	5.27	3460	137	6.17	3650	145	7.1	3860	154	8.05	4060	162	9.05
32400	1600	3130	125	5.06	3350	133	5.99	3565	142	6.92	3765	150	7.91	3960	158	8.9	4160	166	9.94
34425	1700	3270	130	5.78	3475	138	6.79	3680	147	7.77	3885	155	8.78	4055	162	9.88	4250	169	10.93
36450	1800	3410	136	6.75	3607	144	7.68	3810	152	8.725	4010	160	9.8	4180	167	10.1	4350	173	12.1
38475	1900	3546	141	7.52	3730	148	8.67	3935	157	9.71	4120	164	10.90	4320	172	12.0	4455	178	13.1
40500	2000	3700	147	8.54	3860	154	9.71	4050	161	10.83	4255	170	12.1	4423	176	13.2	4580	183	14.4
42525	2100	3850	153	9.60	4000	159	10.85	4210	167	11.97	4350	173	13.3	4535	181	14.6	4680	187	15.8
44550	2200	4000	159	10.74	4168	166	12.17	4320	172	13.40	4500	179	14.7	4670	186	16.0	4800	191	17.3
46575	2300				4323	172	13.44	4450	177	14.70	4623	184	16.1	4770	190	17.5	4930	196	18.9
48600	2400				4460	178	14.55	4620	184	16.30	4740	189	17.7	4920	196	19.2	5045	201	20.5
50625	2500				4600	183	16.40	4720	188	17.90	4880	194	19.4	5036	200	20.8	5170	206	21.3
52650	2600							4910	196	19.80	5000	199	21.3	5180	206	22.8	5325	212	24.3
56700	2800							5180	206	23.4	5280	210	25.2	5435	216	26.5	5410	220	28.6
60750	3000							5485	218	27.8	5610	223	29.7	5650	225	31.3	5840	232	33.1

Volume	Outlet vel.	S. P. 1″			S. P. 1¼″			S. P. 1½″			S. P. 1¾″			S. P. 2″			S. P. 2½″		
		Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.	Tip speed	R.p.m.	B.hp.
24300	1200	3955	158	7.64	4152	166	9.35	4470	178	11.3	4950	197	13.3	5230	208	15.4	5750	229	20.0
26325	1300	4050	161	8.33	4380	175	10.2	4550	182	12.2	5024	200	14.3	5295	211	16.5	5820	232	21.1
28350	1400	4143	165	9.16	4465	178	11.1	4700	187	13.2	5105	203	15.3	5350	213	17.7	5900	235	22.4
30375	1500	4250	169	10.04	4570	182	12.1	4850	193	14.2	5180	206	16.4	5450	217	18.8	5950	237	23.7
32400	1600	4325	172	11.0	4652	186	13.2	4950	197	15.3	5245	209	17.6	5550	222	20.1	6025	240	25.2
34425	1700	4437	177	12.0	4750	189	14.4	5040	200	16.6	5330	212	19.0	5625	224	21.5	6100	243	26.6
36450	1800	4527	180	13.1	4846	193	15.6	5110	203	17.9	5410	216	20.4	5700	227	22.9	6195	247	28.3
38475	1900	4613	184	14.4	4945	197	16.8	5230	208	19.3	5520	220	21.9	5780	230	24.5	6265	249	29.9
40500	2000	4743	189	15.7	5075	202	18.2	5325	212	20.8	5620	224	23.5	5860	233	26.3	6365	253	31.8
42525	2100	4850	193	17.0	5145	205	19.7	5440	216	22.5	5724	228	25.3	5955	237	28.1	6425	257	33.8
44550	2200	4970	198	18.6	5256	209	21.3	5550	221	24.1	5790	230	27.0	6050	241	29.8	6550	261	35.9
46575	2300	5090	203	21.10	5370	214	23.1	5630	224	25.9	5900	235	28.9	6150	245	31.9	6610	263	38.1
48600	2400	5210	208	22.0	5480	218	24.9	5750	229	27.8	6025	240	30.9	6270	250	34.0	6700	267	40.4
50625	2500	5310	213	23.9	5610	224	26.7	5850	233	29.8	6100	243	33.0	6343	252	36.3	6800	271	42.8
52656	2600	5485	218	24.8	5740	229	28.9	5980	238	32.0	6200	247	35.2	6460	257	38.7	6880	274	45.3
56700	2800	5710	228	30.4	5960	238	33.4	6230	248	36.8	6460	257	40.3	6650	265	43.7	7090	282	50.8
60750	3000	5970	238	35.0	6200	247	38.1	6460	257	42.2	6675	265	45.7	6900	274	49.3	7295	290	56.7
64800	3200	6230	252	40.3	6475	258	44.2	6730	268	47.9	6920	276	51.6	7135	284	55.4	7530	300	63.0
68850	3400	6580	262	46.5	6740	268	50.5	6960	277	54.4	7150	285	58.5	7355	293	62.4	7750	308	70.3
72900	3600	6815	272	53.0	7000	279	57.3	7200	287	61.6	7440	296	65.5	7600	303	70.0	8020	320	78.0
76950	3800	7105	283	60.5	7350	292	65.2	7475	297	69.4	7660	305	73.7	7840	312	77.8	8220	328	87.0

STATIC PRESSURE TABLES FOR NIAGARA CONOIDAL FANS[1]

TABLE XIV.—No. 3 NIAGARA CONOIDAL FAN (TYPE N) CAPACITIES AND STATIC PRESSURES AT 70°F. AND 29.92 INCHES BAROMETER

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1/4" S. P.		3/8" S. P.		1/2" S. P.		5/8" S. P.		3/4" S. P.		7/8" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1000	1310	.063	387	.09	483	.15								
1100	1440	.076	384	.11	477	.16								
1200	1570	.090	387	.12	477	.17	557	.23						
1300	1710	.106	393	.14	470	.18	550	.25	623	.32				
1400	1840	.122	400	.16	473	.20	547	.26	617	.33	687	.42		
1500	1970	.141	410	.18	477	.23	543	.28	613	.35	680	.43	743	.52
1600	2100	.160	420	.21	480	.25	547	.31	610	.37	673	.45	733	.54
1700	2230	.180	430	.24	490	.28	550	.34	607	.40	670	.48	727	.56
1800	2360	.202	443	.28	500	.32	553	.37	610	.43	667	.51	723	.59
1900	2490	.225	457	.31	510	.35	560	.41	613	.47	667	.54	720	.62
2000	2630	.250	470	.35	520	.40	570	.45	617	.52	667	.58	720	.66
2100	2760	.275	483	.39	530	.45	580	.50	623	.56	670	.63	720	.71
2200	2890	.302	497	.44	543	.50	590	.55	633	.61	677	.68	723	.76
2300	3020	.330	513	.49	557	.55	600	.61	643	.67	683	.73	727	.81
2400	3150	.360	527	.55	570	.61	610	.67	650	.73	690	.80	733	.87
2500	3280	.390	543	.60	583	.67	623	.74	660	.80	700	.86	740	.94
2600	3410	.422	560	.67	597	.74	633	.81	673	.88	710	.94	747	1.02
2800	3670	.489	590	.81	623	.89	660	.96	693	1.04	730	1.10	767	1.17
3000	3940	.560	623	.99	657	1.04	687	1.14	720	1.22	753	1.29	780	1.36
3200	4190	.638					717	1.33	747	1.42	780	1.50	810	1.58
3400	4460	.721									807	1.75	833	1.84

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1" S. P.		1 1/4" S. P.		1 1/2" S. P.		1 3/4" S. P.		2" S. P.		2 1/4" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1300	1710	.106	820	.58										
1400	1840	.122	810	.59	920	.80	1027	1.00						
1500	1970	.141	800	.62	913	.81	1017	1.04	1110	1.25				
1600	2100	.160	793	.64	903	.84	1007	1.06	1100	1.29	1190	1.53		
1700	2230	.180	783	.66	893	.86	997	1.09	1087	1.32	1177	1.58	1343	2.13
1800	2360	.202	777	.68	883	.89	983	1.12	1077	1.35	1167	1.61	1330	2.16
1900	2490	.225	773	.71	877	.92	977	1.14	1067	1.39	1157	1.65	1317	2.20
2000	2630	.250	770	.75	873	.95	970	1.17	1057	1.42	1143	1.68	1303	2.24
2100	2760	.275	770	.79	867	.99	960	1.22	1050	1.46	1133	1.73	1297	2.29
2200	2890	.302	767	.84	863	1.03	953	1.25	1040	1.50	1127	1.76	1287	2.33
2300	3020	.330	770	.89	860	1.08	950	1.30	1033	1.54	1120	1.81	1270	2.38
2400	3150	.360	773	.95	860	1.13	947	1.35	1027	1.59	1107	1.85	1263	2.43
2500	3280	.390	777	1.03	860	1.20	943	1.41	1023	1.64	1103	1.91	1253	2.49
2600	3410	.422	783	1.09	863	1.26	940	1.47	1020	1.70	1097	1.96	1247	2.54
2800	3670	.489	800	1.25	870	1.43	943	1.63	1013	1.84	1090	2.10	1233	2.67
3000	3940	.560	820	1.44	883	1.61	950	1.81	1020	2.02	1087	2.25	1227	2.82
3200	4190	.638	837	1.65	900	1.83	960	2.02	1023	2.23	1090	2.47	1217	3.00
3400	4460	.721	863	1.90	920	2.06	980	2.26	1033	2.47	1093	2.69	1213	3.21
3600	4730	.810	883	2.18	943	2.34	997	2.53	1050	2.76	1107	2.96	1220	3.48
3800	4990	.900					1017	2.84	1067	3.04	1117	3.28	1227	3.76
4000	5250	1.000							1087	3.39	1133	3.60	1233	4.10

[1] From "Fan Engineering," Buffalo Forge Co.

TABLE XV.—No. 3½ NIAGARA CONOIDAL FAN (TYPE N) CAPACITIES AND STATIC PRESSURES AT 70°F. AND 29.92 INCHES BAROMETER

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	¼" S. P.		⅜" S. P.		½" S. P.		⅝" S. P.		¾" S. P.		⅞" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1000	1790	.063	332	.13	414	.20								
1100	1970	.076	329	.14	409	.21								
1200	2140	.090	332	.16	409	.23	477	.32						
1300	2320	.106	337	.18	403	.25	472	.33	534	.43				
1400	2500	.122	343	.21	406	.28	469	.36	529	.45	589	.57		
1500	2680	.141	352	.24	409	.31	466	.38	526	.48	583	.59	637	.71
1600	2860	.160	360	.28	412	.34	469	.42	523	.51	577	.62	629	.73
1700	3040	.180	369	.32	422	.49	472	.46	520	.55	574	.65	623	.77
1800	3210	.202	380	.37	429	.33	474	.51	523	.59	572	.69	620	.80
1900	3390	.225	392	.42	437	.48	480	.56	526	.64	572	.74	617	.85
2000	3570	.250	403	.48	446	.54	489	.62	529	.70	572	.79	617	.90
2100	3750	.275	414	.53	454	.61	497	.68	534	.76	574	.86	617	.96
2200	3930	.302	426	.59	466	.68	506	.75	543	.83	580	.92	620	1.03
2300	4110	.330	440	.67	477	.75	514	.83	552	.91	586	1.00	623	1.10
2400	4290	.360	452	.74	489	.83	523	.91	557	.99	592	1.09	629	1.18
2500	4470	.390	466	.82	500	.91	534	1.01	566	1.08	600	1.17	634	1.27
2600	4640	.422	480	.91	512	1.01	543	1.10	577	1.19	609	1.27	640	1.39
2800	5000	.489	506	1.10	534	1.21	566	1.31	594	1.41	626	1.50	657	1.59
3000	5360	.560	534	1.35	563	1.42	589	1.56	617	1.65	646	1.75	669	1.85
3200	5720	.638					614	1.81	640	1.94	669	2.05	694	2.16
3400	6070	.721									692	2.38	714	2.50

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1" S. P.		1¼" S. P.		1½" S. P.		1¾" S. P.		2" S. P.		2½" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1300	2320	.106	703	.78										
1400	2500	.122	694	.81	789	1.08	880	1.36						
1500	2680	.141	686	.84	783	1.10	872	1.41	952	1.70				
1600	2860	.160	680	.86	774	1.15	863	1.45	943	1.75	1020	2.08		
1700	3040	.180	672	.89	766	1.17	854	1.48	932	1.79	1009	2.14	1151	2.89
1800	3210	.202	666	.93	757	1.21	843	1.52	923	1.84	1000	2.19	1140	2.94
1900	3390	.225	663	.97	752	1.25	837	1.56	914	1.89	992	2.24	1129	2.99
2000	3570	.250	660	1.02	749	1.30	831	1.59	906	1.94	980	2.29	1117	3.05
2100	3750	.275	660	1.08	743	1.35	823	1.65	900	1.99	972	2.35	1111	3.11
2200	3930	.302	657	1.14	740	1.40	817	1.70	892	2.03	966	2.40	1103	3.17
2300	4110	.330	660	1.22	737	1.47	814	1.77	886	2.10	960	2.46	1089	3.23
2400	4290	.360	663	1.30	737	1.53	812	1.84	880	2.17	949	2.52	1083	3.31
2500	4470	.390	666	1.40	737	1.63	809	1.91	877	2.23	946	2.60	1074	3.38
2600	4640	.422	672	1.48	740	1.72	806	2.00	874	2.32	940	2.67	1069	3.46
2800	5000	.489	686	1.70	746	1.95	809	2.22	869	2.50	934	2.86	1057	3.63
3000	5360	.560	703	1.96	757	2.19	814	2.46	874	2.74	932	3.06	1052	3.84
3200	5720	.638	717	2.24	772	2.49	823	2.75	877	3.04	934	3.36	1043	4.08
3400	6070	.721	740	2.59	789	2.81	840	3.08	886	3.36	937	3.66	1040	4.36
3600	6430	.810	757	2.97	809	3.19	854	3.44	900	3.75	949	4.03	1046	4.73
3800	6790	.900					872	3.86	914	4.14	957	4.46	1052	5.12
4000	7140	1.000							932	4.61	972	4.90	1057	5.59

TABLE XVI.—No. 4 NIAGARA CONOIDAL FAN (TYPE N) CAPACITIES AND STATIC PRESSURES AT 70°F. AND 29.92 INCHES BAROMETER

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	¼" S. P.		⅜" S. P.		½" S. P.		⅝" S. P.		¾" S. P.		⅞" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1000	2330	.063	290	.17	363	.26								
1100	2570	.076	288	.19	358	.28								
1200	2800	.090	290	.21	358	.30	418	.41						
1300	3030	.106	295	.24	353	.33	413	.44	468	.56				
1400	3270	.122	300	.28	355	.36	410	.47	463	.59	515	.74		
1500	3500	.141	308	.32	358	.40	408	.50	460	.62	510	.77	558	.92
1600	3730	.160	315	.37	360	.45	410	.55	458	.66	505	.80	550	.96
1700	3970	.180	323	.42	368	.50	413	.60	455	.71	503	.85	545	1.00
1800	4220	.202	333	.49	375	.56	415	.66	458	.77	500	.90	543	1.05
1900	4430	.225	343	.55	383	.63	420	.73	460	.84	500	.96	540	1.11
2000	4670	.250	353	.62	390	.71	428	.81	463	.92	500	1.04	540	1.17
2100	4900	.275	363	.70	398	.80	435	.89	468	1.00	503	1.12	540	1.26
2200	5130	.302	373	.78	408	.88	443	.98	475	1.08	508	1.21	543	1.35
2300	5370	.330	385	.87	418	.98	450	1.08	483	1.19	513	1.31	545	1.44
2400	5600	.360	395	.97	428	1.09	458	1.19	488	1.30	518	1.42	550	1.55
2500	5830	.390	408	1.07	438	1.19	468	1.32	495	1.41	525	1.53	555	1.67
2600	6070	.422	420	1.19	448	1.32	475	1.43	505	1.56	533	1.67	560	1.81
2800	6530	.489	443	1.44	468	1.58	495	1.71	520	1.84	548	1.95	575	2.08
3000	7000	.560	468	1.76	493	1.86	515	2.03	540	2.16	565	2.29	585	2.42
3200	7460	.638					538	2.37	560	2.53	585	2.67	608	2.82
3400	7930	.721									605	3.11	625	3.27

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1" S. P.		1¼" S. P.		1½" S. P.		1¾" S. P.		2" S. P.		2½" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1300	3030	.106	615	1.03										
1400	3270	.122	608	1.06	690	1.41	770	1.78						
1500	3500	.141	600	1.09	685	1.44	763	1.84	833	2.23				
1600	3730	.160	595	1.13	678	1.50	755	1.89	825	2.29	893	2.72		
1700	3970	.180	588	1.17	670	1.53	748	1.94	815	2.34	883	2.80	1008	3.78
1800	4220	.202	583	1.22	663	1.58	738	1.94	808	2.40	875	2.87	998	3.84
1900	4430	.225	580	1.27	658	1.63	733	2.03	800	2.47	868	2.93	988	3.91
2000	4670	.250	578	1.33	655	1.70	728	2.08	793	2.53	858	2.99	978	3.99
2100	4900	.275	578	1.40	650	1.76	720	2.16	788	2.59	850	3.07	973	4.07
2200	5130	.302	575	1.49	648	1.83	715	2.23	780	2.66	845	3.14	965	4.15
2300	5370	.330	578	1.59	645	1.92	713	2.31	775	2.74	840	3.22	953	4.23
2400	5600	.360	580	1.70	645	2.00	710	2.40	770	2.83	830	3.30	948	4.32
2500	5830	.390	583	1.83	645	2.13	708	2.50	768	2.91	828	3.39	940	4.42
2600	6070	.422	588	1.94	648	2.24	705	2.61	765	3.03	823	3.49	935	4.51
2800	6530	.489	600	2.23	653	2.55	708	2.90	760	3.27	818	3.73	925	4.74
3000	7000	.560	615	2.56	663	2.87	713	3.22	765	3.59	815	4.00	920	5.01
3200	7460	.638	628	2.93	675	3.25	720	3.59	768	3.97	818	4.39	913	5.33
3400	7930	.721	648	3.38	690	3.67	735	4.02	775	4.39	820	4.79	910	5.70
3600	8400	.810	663	3.87	708	4.16	748	4.50	788	4.90	830	5.27	915	6.18
3800	8860	.900					763	5.04	800	5.41	838	5.83	920	6.69
4000	9330	1.000							815	6.02	850	6.40	925	7.30

TABLE XVII.—No. 4½ NIAGARA CONOIDAL FAN (TYPE N) CAPACITIES AND STATIC PRESSURES AT 70°F. AND 29.92 INCHES BAROMETER

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	¼" S. P.		⅜" S. P.		½" S. P.		⅝" S. P.		¾" S. P.		⅞" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1000	2950	.063	258	.21	322	.33								
1100	3250	.076	256	.23	318	.35								
1200	3540	.090	258	.27	318	.38	371	.52						
1300	3840	.106	262	.30	313	.41	367	.55	416	0.71				
1400	4130	.122	267	.35	316	.46	365	.59	411	0.75	458	0.93		
1500	4430	.141	273	.40	318	.51	362	.63	409	0.79	453	0.97	496	1.17
1600	4720	.160	280	.46	320	.57	365	.69	407	0.84	449	1.02	489	1.21
1700	5020	.180	287	.53	327	.64	367	.76	405	0.90	447	1.07	485	1.27
1800	5310	.202	296	.61	333	.71	369	.84	407	0.97	445	1.14	482	1.33
1900	5610	.225	305	.69	340	.80	373	.92	409	1.06	445	1.22	480	1.40
2000	5900	.250	313	.79	347	.89	380	1.02	411	1.16	445	1.31	480	1.48
2100	6200	.275	322	.88	353	1.01	387	1.13	416	1.26	447	1.42	480	1.59
2200	6500	.302	331	.98	362	1.12	393	1.24	422	1.37	451	1.53	482	1.71
2300	6790	.330	342	1.10	371	1.24	400	1.37	429	1.50	456	1.65	485	1.82
2400	7090	.360	351	1.23	380	1.38	407	1.51	433	1.64	460	1.80	489	1.96
2500	7380	.390	362	1.35	389	1.50	416	1.67	440	1.79	467	1.94	493	2.11
2600	7680	.422	373	1.51	398	1.67	422	1.81	449	1.97	473	2.11	498	2.29
2800	8270	.489	393	1.82	416	2.00	440	2.17	462	2.33	487	2.47	511	2.63
3000	8860	.560	416	2.23	438	2.35	458	2.57	480	2.73	502	2.90	520	3.06
3200	9450	.638					478	3.00	498	3.20	520	3.38	540	3.57
3400.	10040	.721									538	3.93	556	4.13

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1" S. P.		1¼" S. P.		1½" S. P.		1¾" S. P.		2" S. P.		2½" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1300	3840	.106	547	1.30										
1400	4130	.122	540	1.34	613	1.79	685	2.25						
1500	4430	.141	533	1.38	609	1.82	678	2.33	740	2.82				
1600	4720	.160	529	1.43	602	1.89	671	2.39	733	2.90	793	3.44		
1700	5020	.180	522	1.48	596	1.93	665	2.45	725	2.96	785	3.54	896	4.78
1800	5310	.202	518	1.54	589	2.00	656	2.51	718	3.04	778	3.63	887	4.86
1900	5610	.225	516	1.60	585	2.07	651	2.57	711	3.12	771	3.71	878	4.94
2000	5900	.250	513	1.69	582	2.15	647	2.63	704	3.20	762	3.79	869	5.04
2100	6200	.275	513	1.78	578	2.23	640	2.74	700	3.28	756	3.89	865	5.14
2200	6500	.302	511	1.89	576	2.31	636	2.82	696	3.36	751	3.97	858	5.25
2300	6790	.330	513	2.01	573	2.43	633	2.92	689	3.46	747	4.07	847	5.35
2400	7090	.360	516	2.15	573	2.53	631	3.04	685	3.59	738	4.17	842	5.47
2500	7380	.390	518	2.31	573	2.69	629	3.16	682	3.69	736	4.29	836	5.59
2600	7680	.422	522	2.45	576	2.84	627	3.30	680	3.83	731	4.42	831	5.71
2800	8270	.489	533	2.82	580	3.22	629	3.67	676	4.13	727	4.72	822	5.99
3000	8860	.560	547	3.24	589	3.63	633	4.07	680	4.54	725	5.06	818	6.34
3200	9450	.638	558	3.71	600	4.11	640	4.54	682	5.02	727	5.55	811	6.74
3400	10040	.721	576	4.27	613	4.64	653	5.08	689	5.55	729	6.06	809	7.21
3600	10630	.810	589	4.90	629	5.27	665	5.69	700	6.20	738	6.66	813	7.82
3800	11220	.900					678	6.38	711	6.85	745	7.37	818	8.46
4000	11810	1.000							725	7.61	756	8.10	822	9.23

TABLE XVIII.—No. 5 NIAGARA CONOIDAL FAN (TYPE N) CAPACITIES AND STATIC PRESSURES AT 70°F. AND 29.92 INCHES BAROMETER

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	¼" S. P.		⅜" S. P.		½" S. P.		⅝" S. P.		¾" S. P.		⅞" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1000	3640	.063	232	.26	290	.41								
1100	4010	.076	230	.29	286	.44								
1200	4370	.090	232	.33	286	.47	334	.65						
1300	4740	.106	236	.38	282	.51	330	.68	374	.88				
1400	5100	.122	240	.43	284	.56	328	.73	370	.92	412	1.15		
1500	5470	.141	246	.50	286	.63	326	.78	368	.98	408	1.20	446	1.44
1600	5830	.160	252	.57	288	.70	328	.86	366	1.04	404	1.26	440	1.49
1700	6190	.180	258	.66	294	.79	330	.94	364	1.11	402	1.33	436	1.57
1800	6560	.202	266	.76	300	.88	332	1.03	366	1.20	400	1.40	434	1.64
1900	6930	.225	274	.86	306	.99	336	1.14	368	1.31	400	1.50	432	1.73
2000	7290	.250	282	.97	312	1.11	342	1.26	370	1.43	400	1.62	432	1.83
2100	7660	.275	290	1.09	318	1.24	348	1.39	374	1.56	402	1.75	432	1.96
2200	8010	.302	298	1.21	326	1.38	354	1.53	380	1.69	406	1.89	434	2.11
2300	8380	.330	308	1.36	334	1.55	360	1.69	386	1.85	410	2.04	436	2.25
2400	8750	.360	316	1.51	342	1.70	366	1.86	390	2.03	414	2.22	440	2.41
2500	9100	.390	326	1.67	350	1.86	374	2.06	396	2.21	420	2.40	444	2.60
2600	9480	.422	336	1.86	358	2.06	380	2.24	404	2.43	426	2.60	448	2.83
2800	10200	.489	354	2.25	374	2.46	396	2.68	416	2.88	438	3.05	460	3.25
3000	10940	.560	374	2.75	394	2.90	412	3.18	432	3.38	452	3.58	468	3.78
3200	11660	.638					430	3.70	448	3.95	468	4.18	486	4.40
3400	12390	.721									484	4.85	500	5.10

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1" S. P.		1¼" S. P.		1½" S. P.		1¾" S. P.		2" S. P.		2½" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1300	4740	.106	492	1.60										
1400	5100	.122	486	1.65	552	2.21	616	2.78						
1500	5470	.141	480	1.71	548	2.25	610	2.88	666	3.48				
1600	5830	.160	476	1.76	542	2.34	604	2.95	660	3.58	714	4.25		
1700	6190	.180	470	1.82	536	2.39	598	3.03	652	3.65	706	4.38	806	5.90
1800	6560	.202	466	1.90	530	2.47	590	3.10	646	3.75	700	4.48	798	6.00
1900	6930	.225	464	1.98	526	2.55	586	3.18	640	3.85	694	4.58	790	6.10
2000	7290	.250	462	2.08	524	2.65	582	3.25	634	3.95	686	4.68	782	6.23
2100	7660	.275	462	2.19	520	2.75	576	3.38	630	4.05	680	4.80	778	6.35
2200	8010	.302	460	2.33	518	2.85	572	3.48	624	4.15	676	4.90	772	6.48
2300	8380	.330	462	2.48	516	3.00	570	3.60	620	4.28	672	5.03	762	6.60
2400	8750	.360	464	2.65	516	3.13	568	3.75	616	4.44	664	5.15	758	6.75
2500	9100	.390	466	2.85	516	3.33	566	3.90	614	4.55	662	5.30	752	6.90
2600	9480	.422	470	3.03	518	3.50	564	4.08	612	4.73	658	5.45	748	7.05
2800	10200	.489	480	3.48	522	3.98	566	4.53	608	5.10	654	5.83	740	7.40
3000	10940	.560	492	4.00	530	4.48	570	5.03	612	5.60	652	6.25	736	7.83
3200	11660	.638	502	4.57	540	5.08	576	5.60	614	6.20	654	6.85	730	8.32
3400	12390	.721	518	5.27	552	5.73	588	6.28	620	6.85	656	7.48	728	8.90
3600	13120	.810	530	6.05	566	6.50	598	7.03	630	7.65	664	8.22	732	9.65
3800	13850	.900					610	7.88	640	8.46	670	9.10	736	10.5
4000	14580	1.000							652	9.40	680	10.0	740	11.4

TABLE XIX.—No. 5½ NIAGARA CONOIDAL FAN (TYPE N) CAPACITIES AND STATIC PRESSURES AT 70°F. AND 29.92 INCHES BAROMETER

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	¼" S. P.		⅜" S. P.		½" S. P.		⅝" S. P.		¾" S. P.		⅞" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1000	4410	.063	211	.32	264	.49								
1100	4850	.076	209	.35	260	.53								
1200	5290	.090	211	.40	260	.57	304	.78						
1300	5730	.106	215	.45	257	.62	300	.83	340	1.06				
1400	6170	.122	218	.52	258	.68	298	.88	336	1.12	375	1.40		
1500	6620	.141	224	.60	260	.76	296	.95	335	1.18	371	1.45	406	1.75
1600	7060	.160	229	.69	262	.85	298	1.04	333	1.26	367	1.52	400	1.81
1700	7500	.180	235	.80	267	.95	300	1.13	331	1.35	366	1.60	397	1.89
1800	7940	.202	242	.92	273	1.06	302	1.25	333	1.46	364	1.70	395	1.98
1900	8380	.225	249	1.04	278	1.19	306	1.38	335	1.59	364	1.82	393	2.09
2000	8820	.250	256	1.17	284	1.34	311	1.53	336	1.73	364	1.96	393	2.21
2100	9260	.275	264	1.32	289	1.50	316	1.68	340	1.88	366	2.12	393	2.37
2200	9700	.302	271	1.47	296	1.67	322	1.85	346	2.05	369	2.28	395	2.55
2300	10140	.330	280	1.65	304	1.86	327	2.05	351	2.24	373	2.47	397	2.72
2400	10590	.360	287	1.83	311	2.05	333	2.25	355	2.45	377	2.68	400	2.92
2500	11030	.390	297	2.02	318	2.25	340	2.49	360	2.67	382	2.90	404	3.15
2600	11470	.422	306	2.25	326	2.49	346	2.71	367	2.94	387	3.15	407	3.42
2800	12350	.489	322	2.72	340	2.98	360	3.24	378	3.48	398	3.69	418	3.93
3000	13230	.560	340	3.33	358	3.51	375	3.84	393	4.08	411	4.33	426	4.57
3200	14110	.638					391	4.48	407	4.78	426	5.05	442	5.33
3400	15000	.721									440	5.87	455	6.17

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1" S. P.		1¼" S. P.		1½" S. P.		1¾" S. P.		2" S. P.		2½" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1300	5730	.106	447	1.94										
1400	6170	.122	442	1.99	502	2.67	560	3.36						
1500	6620	.141	437	2.07	498	2.72	555	3.48	606	4.21				
1600	7060	.160	433	2.13	493	2.83	549	3.57	600	4.33	649	5.14		
1700	7500	.180	427	2.20	487	2.89	544	3.66	593	4.42	642	5.29	733	7.14
1800	7940	.202	424	2.30	482	2.99	537	3.75	587	4.54	636	5.42	726	7.26
1900	8380	.225	422	2.39	478	3.09	533	3.84	582	4.66	631	5.54	718	7.38
2000	8820	.250	420	2.52	476	3.21	529	3.93	576	4.78	624	5.66	711	7.53
2100	9260	.275	420	2.65	473	3.33	524	4.08	573	4.90	618	5.81	707	7.68
2200	9700	.302	418	2.82	471	3.45	520	4.21	567	5.02	615	5.93	702	7.84
2300	10140	.330	420	3.00	469	3.63	518	4.36	564	5.17	611	6.08	693	7.99
2400	10590	.360	422	3.21	469	3.78	517	4.54	560	5.35	604	6.23	689	8.17
2500	11030	.390	424	3.45	469	4.02	515	4.72	558	5.51	602	6.41	684	8.35
2600	11470	.422	427	3.66	471	4.24	513	4.93	557	5.72	598	6.59	680	8.53
2800	12350	.489	437	4.21	475	4.81	515	5.48	553	6.17	595	7.05	673	8.95
3000	13230	.560	447	4.84	482	5.42	518	6.08	557	6.78	593	7.56	669	9.47
3200	14110	.638	456	5.54	491	6.14	524	6.78	558	7.50	595	8.29	664	10.1
3400	15000	.721	471	6.38	502	6.93	535	7.59	564	8.29	596	9.04	662	10.8
3600	15880	.810	482	7.32	515	7.87	544	8.50	573	9.26	604	9.95	666	11.7
3800	16760	.900					555	9.53	582	10.2	609	11.0	669	12.7
4000	17640	1.000							593	11.4	618	12.1	673	13.8

Table XX.—No. 6 Niagara Conoidal Fan (Type N) Capacities and Static Pressures at 70°F. and 29.92 Inches Barometer

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	¼" S. P.		⅜" S. P.		½" S. P.		⅝" S. P.		¾" S. P.		⅞" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1000	5250	.063	193	.37	242	.59								
1100	5770	.076	192	.42	238	.63								
1200	6300	.090	193	.48	238	.67	278	.93						
1300	6820	.106	197	.54	235	.73	275	.98	312	1.27				
1400	7350	.122	200	.62	237	.81	274	1.05	308	1.33	344	1.66		
1500	7870	.141	205	.72	238	.91	272	1.13	307	1.41	340	1.72	372	2.08
1600	8400	.160	210	.82	240	1.01	274	1.23	305	1.49	337	1.81	367	2.15
1700	8920	.180	215	.95	245	1.13	275	1.35	304	1.60	335	1.91	363	2.25
1800	9450	.202	222	1.09	250	1.26	277	1.49	305	1.73	334	2.02	362	2.36
1900	9970	.225	228	1.24	255	1.42	280	1.64	307	1.88	334	2.16	360	2.49
2000	10500	.250	235	1.40	260	1.59	285	1.82	309	2.06	334	2.33	360	2.63
2100	11030	.275	242	1.57	265	1.79	290	2.00	312	2.24	335	2.52	360	2.82
2200	11550	.302	248	1.75	272	1.98	295	2.20	317	2.43	339	2.72	362	3.04
2300	12070	.330	257	1.96	279	2.21	300	2.43	322	2.66	342	2.94	363	3.23
2400	12600	.360	263	2.18	285	2.45	305	2.68	325	2.92	345	3.19	367	3.48
2500	13120	.390	272	2.41	291	2.67	312	2.96	330	3.18	350	3.45	370	3.74
2600	13650	.422	280	2.68	299	2.96	317	3.22	337	3.50	355	3.74	374	4.07
2800	14700	.489	295	3.24	312	3.55	330	3.85	347	4.14	365	4.39	384	4.68
3000	15750	.560	312	3.96	329	4.18	344	4.57	360	4.86	377	5.15	390	5.44
3200	16790	.638					359	5.33	373	5.69	390	6.01	405	6.34
3400	17850	.721									403	6.98	417	7.35

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1" S. P.		1¼" S. P.		1½" S. P.		1¾" S. P.		2" S. P.		2½" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1300	6820	.106	410	2.31										
1400	7350	.122	405	2.37	460	3.18	513	4.00						
1500	7870	.141	400	2.46	457	3.24	509	4.14	555	5.00				
1600	8400	.160	397	2.54	452	3.36	504	4.25	550	5.15	595	6.12		
1700	8920	.180	392	2.62	447	3.44	499	4.36	544	5.26	589	6.30	672	8.50
1800	9450	.202	389	2.73	442	3.56	492	4.47	539	5.40	584	6.45	665	8.64
1900	9970	.225	387	2.85	439	3.67	489	4.57	534	5.55	579	6.59	659	8.78
2000	10500	.250	385	3.00	437	3.82	485	4.68	529	5.69	572	6.73	652	8.96
2100	11030	.275	385	3.16	434	3.96	480	4.86	525	5.83	567	6.91	649	9.14
2200	11550	.302	384	3.35	432	4.11	477	5.00	520	5.98	564	7.06	644	9.32
2300	12070	.330	385	3.57	430	4.32	475	5.18	517	6.16	560	7.24	635	9.50
2400	12600	.360	387	3.82	430	4.50	474	5.40	514	6.37	554	7.42	632	9.72
2500	13120	.390	389	4.10	430	4.79	472	5.62	512	6.55	552	7.63	627	9.94
2600	13650	.422	392	4.36	432	5.04	470	5.87	510	6.81	549	7.85	624	10.2
2800	14700	.489	400	5.00	435	5.73	472	6.52	507	7.34	545	8.39	617	10.7
3000	15750	.560	410	5.76	442	6.45	475	7.24	510	8.06	544	9.00	614	11.3
3200	16790	.638	419	6.59	450	7.31	480	8.06	512	8.93	545	9.86	609	12.0
3400	17850	.721	432	7.60	460	8.24	490	9.04	517	9.86	547	10.8	607	12.8
3600	18900	.810	442	8.71	472	9.36	499	10.1	525	11.0	554	11.9	610	13.9
3800	19950	.900					509	11.3	534	12.2	559	13.1	614	15.1
4000	21000	1.000							544	13.5	567	14.4	617	16.4

TABLE XXI.—No. 7 NIAGARA CONOIDAL FAN (TYPE N) CAPACITIES AND STATIC PRESSURES AT 70°F. AND 29.92 INCHES BAROMETER

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	¼" S. P.		⅜" S. P.		½" S. P.		⅝" S. P.		¾" S. P.		⅞" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1000	7140	.063	166	.51	207	.80								
1100	7860	.076	164	.57	204	.85								
1200	8570	.090	166	.65	204	.92	239	1.26						
1300	9290	.106	169	.74	202	1.00	236	1.34	267	1.73				
1400	10000	.122	172	.85	203	1.10	234	1.43	264	1.81	294	2.26		
1500	10720	.141	176	.98	204	1.24	283	1.53	263	1.91	292	2.34	319	2.83
1600	11430	.160	180	1.12	206	1.37	234	1.68	262	2.03	289	2.46	314	2.93
1700	12150	.180	184	1.29	210	1.54	236	1.83	260	2.18	287	2.60	312	3.07
1800	12860	.202	190	1.49	214	1.72	237	2.02	262	2.36	286	2.75	310	3.21
1900	13570	.225	196	1.68	219	1.93	240	2.23	263	2.56	286	2.95	309	3.39
2000	14290	.250	202	1.90	223	2.17	244	2.47	264	2.80	286	3.18	309	3.58
2100	15000	.275	207	2.13	227	2.44	249	2.73	267	3.05	287	3.43	309	3.84
2200	15720	.302	213	2.38	233	2.70	253	3.00	272	3.31	290	3.70	310	4.13
2300	16430	.330	220	2.67	239	3.01	257	3.31	276	3.63	293	4.00	312	4.40
2400	17150	.360	226	2.97	244	3.33	262	3.64	279	3.97	296	4.34	314	4.73
2500	17860	.390	233	3.27	250	3.64	267	4.03	283	4.33	300	4.70	317	5.10
2600	18580	.422	240	3.64	256	4.03	272	4.39	289	4.77	304	5.10	320	5.54
2800	20000	.489	253	4.41	267	4.83	283	5.24	297	5.64	313	5.98	329	6.37
3000	21430	.560	267	5.39	282	5.68	294	6.22	309	6.62	323	7.01	334	7.40
3200	22860	.638					307	7.25	320	7.74	334	8.18	347	8.62
3400	24290	.721									346	9.51	357	10.0

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1" S. P.		1¼" S. P.		1½" S. P.		1¾" S. P.		2" S. P.		2½" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1300	9290	.106	352	3.14										
1400	10000	.122	347	3.23	394	4.33	440	5.44						
1500	10720	.141	343	3.35	392	4.41	436	5.64	476	6.81				
1600	11430	.160	340	3.46	387	4.58	432	5.78	472	7.01	510	8.33		
1700	12150	.180	336	3.57	383	4.68	427	5.93	466	7.15	504	8.58	576	11.6
1800	12860	.202	333	3.72	379	4.85	422	6.08	462	7.35	500	8.77	570	11.8
1900	13570	.225	332	3.88	376	5.00	419	6.22	457	7.55	496	8.97	564	12.0
2000	14290	.250	330	4.08	374	5.19	416	6.37	453	7.74	490	9.16	559	12.2
2100	15000	.275	330	4.30	372	5.39	412	6.62	450	7.94	486	9.41	556	12.5
2200	15720	.302	329	4.56	370	5.59	409	6.81	446	8.13	483	9.60	552	12.7
2300	16430	.330	330	4.86	369	5.88	407	7.06	443	8.38	480	9.85	544	12.9
2400	17150	.360	332	5.19	369	6.13	406	7.35	440	8.67	474	10.1	542	13.2
2500	17860	.390	333	5.59	369	6.52	404	7.64	439	8.92	473	10.4	537	13.5
2600	18580	.422	336	5.93	370	6.86	403	7.99	437	9.26	470	10.7	534	13.8
2800	20000	.489	343	6.81	373	7.79	404	8.87	434	10.0	467	11.4	529	14.5
3000	21430	.560	352	7.84	379	8.77	407	9.85	437	11.0	466	12.3	526	15.3
3200	22860	.638	359	8.97	386	9.95	412	11.0	439	12.2	467	13.4	522	16.3
3400	24290	.721	370	10.3	394	11.2	420	12.3	443	13.4	469	14.7	520	17.4
3600	25720	.810	379	11.9	404	12.7	427	13.8	450	15.0	474	16.1	523	18.9
3800	27150	.900					436	15.4	457	16.6	479	17.8	526	20.5
4000	28580	1.000							466	18.4	486	19.6	529	22.4

Table XXII.—No. 8 Niagara Conoidal Fan (Type N) Capacities and Static Pressures at 70°F. and 29.92 Inches Barometer

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1/4" S. P.		3/8" S. P.		1/2" S. P.		5/8" S. P.		3/4" S. P.		7/8" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1000	9330	.063	145	.67	181	1.04								
1100	10270	.076	144	.74	179	1.11								
1200	11200	.090	145	.85	179	1.20	209	1.65						
1300	12130	.106	148	.96	176	1.31	206	1.75	234	2.25				
1400	13060	.122	150	1.11	178	1.44	205	1.87	231	2.36	258	2.95		
1500	14000	.141	154	1.27	179	1.61	204	2.00	230	2.50	255	3.06	279	3.69
1600	14930	.160	158	1.47	180	1.79	205	2.19	229	2.66	253	3.21	275	3.82
1700	15860	.180	161	1.69	184	2.01	206	2.39	228	2.85	251	3.39	273	4.01
1800	16800	.202	166	1.94	188	2.25	208	2.64	229	3.08	250	3.59	271	4.19
1900	17730	.225	171	2.20	191	2.52	210	2.91	230	3.34	250	3.85	270	4.42
2000	18660	.250	176	2.48	195	2.83	214	3.23	231	3.66	250	4.15	270	4.68
2100	19600	.275	181	2.79	199	3.18	218	3.56	234	3.98	251	4.48	270	5.02
2200	20530	.302	186	3.11	204	3.53	221	3.92	238	4.33	254	4.83	271	5.40
2300	21460	.330	193	3.48	209	3.93	225	4.33	241	4.74	256	5.22	273	5.75
2400	22400	.360	198	3.87	214	4.35	229	4.76	244	5.19	259	5.67	275	6.18
2500	23330	.390	204	4.28	219	4.75	234	5.26	248	5.65	263	6.13	278	6.66
2600	24260	.422	210	4.76	224	5.26	238	5.73	253	6.23	266	6.66	280	7.23
2800	26130	.489	221	5.76	234	6.31	248	6.85	260	7.36	274	7.81	288	8.32
3000	28000	.560	234	7.04	246	7.42	258	8.13	270	8.64	283	9.15	293	9.66
3200	29860	.638					269	9.47	280	10.1	293	10.7	304	11.3
3400	31720	.721									303	12.4	313	13.1

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1" S. P.		1 1/4" S. P.		1 1/2" S. P.		1 3/4" S. P.		2" S. P.		2 1/2" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1300	12130	.106	308	4.10										
1400	13060	.122	304	4.22	345	5.65	385	7.10						
1500	14000	.141	300	4.37	343	5.76	381	7.36	416	8.90				
1600	14930	.160	298	4.51	339	5.98	378	7.55	413	9.15	446	10.9		
1700	15860	.180	294	4.66	335	6.11	374	7.74	408	9.34	441	11.2	504	15.1
1800	16800	.202	291	4.86	331	6.33	369	7.94	404	9.60	438	11.5	499	15.4
1900	17730	.225	290	5.06	329	6.53	366	8.13	400	9.86	434	11.7	494	15.6
2000	18660	.250	289	5.33	328	6.78	364	8.32	396	10.1	429	12.0	489	15.9
2100	19600	.275	289	5.61	325	7.04	360	8.64	394	10.4	425	12.3	486	16.3
2200	20530	.302	288	5.96	324	7.30	358	8.90	390	10.6	423	12.6	483	16.6
2300	21460	.330	289	6.35	323	7.68	356	9.22	388	11.0	420	12.9	476	16.9
2400	22400	.360	290	6.78	323	8.00	355	9.60	385	11.3	415	13.2	474	17.3
2500	23330	.390	291	7.30	323	8.51	354	9.98	384	11.7	414	13.6	470	17.7
2600	24260	.422	294	7.74	324	8.96	353	10.4	383	12.1	411	14.0	468	18.1
2800	26130	.489	300	8.90	326	10.2	354	11.6	380	13.1	409	14.9	463	19.0
3000	28000	.560	308	10.2	331	11.5	356	12.9	383	14.3	408	16.0	460	20.0
3200	29860	.638	314	11.7	338	13.0	360	14.3	384	15.9	409	17.5	456	21.3
3400	31720	.721	324	13.5	345	14.7	368	16.1	388	17.5	410	19.1	455	22.8
3600	33590	.810	331	15.5	354	16.6	374	18.0	394	19.6	415	21.1	458	24.7
3800	35460	.900					381	20.2	400	21.6	419	23.3	460	26.8
4000	37330	1.000							408	24.1	425	25.6	463	29.2

TABLE XXIII.—NO. 9 NIAGARA CONOIDAL FAN (TYPE N) CAPACITIES AND STATIC PRESSURES AT 70°F. AND 29.92 INCHES BAROMETER

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	¼" S. P.		⅜" S. P.		½" S. P.		⅝" S. P.		¾" S. P.		⅞" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1000	11810	.063	129	.84	161	1.32								
1100	12990	.076	128	.94	159	1.41								
1200	14170	.090	129	1.07	159	1.52	186	2.09						
1300	15360	.106	131	1.22	157	1.65	183	2.21	208	2.85				
1400	16530	.122	133	1.40	158	1.82	182	2.37	206	2.99	229	3.74		
1500	17720	.141	137	1.61	159	2.04	181	2.54	205	3.16	227	3.87	248	4.67
1600	18900	.160	140	1.86	160	2.27	182	2.77	203	3.36	225	4.07	244	4.84
1700	20080	.180	143	2.14	163	2.54	183	3.03	202	3.60	223	4.29	242	5.07
1800	21250	.202	148	2.45	167	2.84	185	3.35	203	3.90	222	4.55	241	5.30
1900	22440	.225	152	2.78	170	3.19	187	3.69	205	4.23	222	4.87	240	5.60
2000	23620	.250	157	3.14	173	3.58	190	4.08	206	4.64	222	5.25	240	5.92
2100	24800	.275	161	3.52	177	4.03	193	4.51	208	5.04	223	5.67	240	6.35
2200	25980	.302	166	3.93	181	4.47	197	4.96	211	5.47	226	6.10	241	6.83
2300	27160	.330	171	4.41	186	4.97	200	5.48	215	6.00	228	6.61	242	7.27
2400	28340	.360	176	4.90	190	5.50	203	6.02	217	6.56	230	7.18	244	7.82
2500	29520	.390	181	5.41	195	6.01	208	6.66	220	7.15	233	7.76	247	8.43
2600	30710	.422	187	6.02	199	6.66	211	7.25	224	7.88	237	8.42	249	9.15
2800	33070	.489	197	7.28	208	7.98	220	8.67	231	9.30	243	9.88	256	10.5
3000	35430	.560	208	8.91	219	9.40	229	10.3	240	10.9	251	11.6	260	12.2
3200	37790	.638					239	12.0	249	12.8	260	13.5	270	14.3
3400	40150	.721									269	15.7	278	16.5

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1" S. P.		1¼" S. P.		1½" S. P.		1¾" S. P.		2" S. P.		2¼" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1300	15360	.106	273	5.18										
1400	16530	.122	270	5.34	307	7.15	342	8.99						
1500	17720	.141	267	5.53	304	7.29	339	9.31	370	11.3				
1600	18900	.160	264	5.71	301	7.57	336	9.56	367	11.6	397	13.8		
1700	20080	.180	261	5.90	298	7.73	332	9.80	362	11.8	392	14.2	448	19.1
1800	21250	.202	259	6.15	294	8.01	328	10.0	359	12.2	389	14.5	443	19.4
1900	22440	.225	258	6.41	292	8.26	326	10.3	356	12.5	386	14.8	439	19.8
2000	23620	.250	257	6.74	291	8.59	323	10.5	352	12.8	381	15.2	435	20.2
2100	24800	.275	257	7.10	289	8.91	320	10.9	350	13.1	378	15.6	432	20.6
2200	25980	.302	256	7.54	288	9.23	318	11.3	347	13.4	376	15.9	429	21.0
2300	27160	.330	257	8.04	287	9.72	317	11.7	344	13.7	373	16.3	423	21.4
2400	28340	.360	258	8.59	287	10.1	316	12.2	342	14.3	369	16.7	421	21.9
2500	29520	.390	259	9.23	287	10.8	314	12.6	341	14.8	368	17.2	418	22.4
2600	30710	.422	261	9.80	288	11.3	313	13.2	340	15.3	366	17.7	416	22.8
2800	33070	.489	267	11.3	290	12.9	314	14.7	338	16.5	363	18.9	411	24.0
3000	35430	.560	273	13.0	294	14.5	317	16.3	340	18.2	362	20.3	409	25.4
3200	37790	.638	279	14.8	300	16.4	320	18.1	341	20.1	363	22.2	406	27.0
3400	40150	.721	288	17.1	307	18.6	327	20.3	344	22.2	364	24.2	405	28.8
3600	42510	.810	294	19.6	314	21.1	332	22.8	350	24.8	369	26.7	407	31.3
3800	44880	.900					339	25.5	356	27.4	372	29.5	409	33.9
4000	47240	1.000							362	30.5	378	32.4	411	36.9

TABLE XXIV.—No. 10 NIAGARA CONOIDAL FAN (TYPE N) CAPACITIES AND STATIC PRESSURES AT 70°F. AND 29.92 INCHES BAROMETER

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1/4" S. P.		3/8" S. P.		1/2" S. P.		5/8" S. P.		3/4" S. P.		7/8" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1000	14580	.063	116	1.04	145	1.63								
1100	16040	.076	115	1.16	143	1.74								
1200	17500	.090	116	1.32	143	1.87	167	2.58						
1300	18960	.106	118	1.50	141	2.04	165	2.73	187	3.52				
1400	20410	.122	120	1.73	142	2.25	164	2.92	185	3.69	206	4.61		
1500	21870	.141	123	1.99	143	2.52	163	3.13	184	3.90	204	4.78	223	5.77
1600	23330	.160	126	2.29	144	2.80	164	3.42	183	4.15	202	5.02	220	5.97
1700	24790	.180	129	2.64	147	3.14	165	3.74	182	4.45	201	5.30	218	6.26
1800	26240	.202	133	3.03	150	3.51	166	4.13	183	4.81	200	5.61	217	6.55
1900	27700	.225	137	3.43	153	3.94	168	4.55	184	5.22	200	6.01	216	6.91
2000	29160	.250	141	3.88	156	4.42	171	5.04	185	5.72	200	6.48	216	7.31
2100	30620	.275	145	4.35	159	4.97	174	5.56	187	6.22	201	7.00	216	7.84
2200	32080	.302	149	4.85	163	5.51	177	6.12	190	6.76	203	7.54	217	8.43
2300	33540	.330	154	5.44	167	6.14	180	6.76	193	7.40	205	8.16	218	8.98
2400	34990	.360	158	6.05	171	6.79	183	7.43	195	8.10	207	8.86	220	9.65
2500	36450	.390	163	6.68	175	7.42	187	8.22	198	8.83	210	9.58	222	10.4
2600	37910	.422	168	7.43	179	8.22	190	8.95	202	9.73	213	10.4	224	11.3
2800	40830	.489	177	8.99	187	9.85	198	10.7	208	11.5	219	12.2	230	13.0
3000	43740	.560	187	11.0	197	11.6	206	12.7	216	13.5	226	14.3	234	15.1
3200	46660	.638					215	14.8	224	15.8	234	16.7	243	17.6
3400	49570	.721									242	19.4	250	20.4

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1" S. P.		1 1/4" S. P.		1 1/2" S. P.		1 3/4" S. P.		2" S. P.		2 1/2" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1300	18960	.106	246	6.40										
1400	20410	.122	243	6.59	276	8.83	308	11.1						
1500	21870	.141	240	6.83	274	9.00	305	11.5	333	13.9				
1600	23330	.160	238	7.05	271	9.34	302	11.8	330	14.3	357	17.0		
1700	24790	.180	235	7.28	268	9.54	299	12.1	326	14.6	353	17.5	403	23.6
1800	26240	.202	233	7.59	265	9.89	295	12.4	323	15.0	350	17.9	399	24.0
1900	27700	.225	232	7.91	263	10.2	293	12.7	320	15.4	347	18.3	395	24.4
2000	29160	.250	231	8.32	262	10.6	291	13.0	317	15.8	343	18.7	391	24.9
2100	30620	.275	231	8.77	260	11.0	288	13.5	315	16.2	340	19.2	389	25.4
2200	32080	.302	230	9.31	259	11.4	286	13.9	312	16.6	338	19.6	386	25.9
2300	33540	.330	231	9.92	258	12.0	285	14.4	310	17.1	336	20.1	381	26.4
2400	34990	.360	232	10.6	258	12.5	284	15.0	308	17.7	332	20.6	379	27.0
2500	36450	.390	233	11.4	258	13.3	283	15.6	307	18.2	331	21.2	376	27.6
2600	37910	.422	235	12.1	259	14.0	282	16.3	306	18.9	329	21.8	374	28.2
2800	40830	.489	240	13.9	261	15.9	283	18.1	304	20.4	327	23.3	370	29.6
3000	43740	.560	246	16.0	265	17.9	285	20.1	306	22.4	326	25.0	368	31.3
3200	46660	.638	251	18.3	270	20.3	288	22.4	307	24.8	327	27.4	365	33.3
3400	49570	.721	259	21.1	276	22.9	294	25.1	310	27.4	328	29.9	364	35.6
3600	52490	.810	265	24.2	283	26.0	299	28.1	315	30.6	332	32.9	366	38.6
3800	55400	.900					305	31.5	320	33.8	335	36.4	368	41.8
4000	58320	1.000							326	37.6	340	40.0	370	45.6

TABLE XXV.—No. 11 NIAGARA CONOIDAL FAN (TYPE N) CAPACITIES AND STATIC PRESSURES AT 70°F. AND 29.92 INCHES BAROMETER

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1/4" S. P.		3/8" S. P.		1/2" S. P.		5/8" S. P.		3/4" S. P.		7/8" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1000	17640	.063	106	1.26	132	1.97								
1100	19410	.076	105	1.40	130	2.11								
1200	21170	.090	106	1.60	130	2.26	152	3.12						
1300	22930	.106	107	1.82	128	2.47	150	3.30	170	4.26				
1400	24700	.122	109	2.09	129	2.72	149	3.53	168	4.47	187	5.58		
1500	26460	.141	112	2.41	130	3.05	148	3.79	167	4.72	186	5.78	203	6.98
1600	28230	.160	115	2.77	131	3.39	149	4.14	166	5.02	184	6.08	200	7.22
1700	29990	.180	117	3.20	134	3.80	150	4.53	166	5.39	183	6.41	198	7.58
1800	31750	.202	121	3.67	136	4.25	151	5.00	166	5.82	182	6.79	197	7.93
1900	33520	.225	125	4.15	139	4.77	153	5.51	167	6.32	182	7.27	196	8.36
2000	35280	.250	128	4.70	142	5.35	156	6.10	168	6.92	182	7.84	196	8.85
2100	37050	.275	132	5.26	145	6.01	158	6.73	170	7.53	183	8.87	196	9.49
2200	38810	.302	136	5.87	148	6.67	161	7.41	173	8.18	185	9.12	197	10.2
2300	40580	.330	140	6.58	152	7.43	164	8.18	176	8.95	186	9.87	198	10.9
2400	42340	.360	144	7.32	156	8.22	166	8.99	177	9.80	188	10.7	200	11.7
2500	44100	.390	148	8.08	159	8.98	170	9.95	180	10.7	191	11.6	202	12.6
2600	45870	.422	153	8.99	163	9.95	173	10.8	184	11.8	194	12.6	204	13.7
2800	49400	.489	161	10.9	170	11.9	180	13.0	189	13.9	199	14.8	209	15.7
3000	52910	.560	170	13.3	179	14.0	187	15.4	196	16.3	206	17.3	213	18.3
3200	56450	.638					196	17.9	204	19.1	213	20.2	221	21.3
3400	59980	.721									220	23.5	227	24.7

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1" S. P.		1 1/4" S. P.		1 1/2" S. P.		1 3/4" S. P.		2" S. P.		2 1/4" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1300	22930	.106	224	7.74										
1400	24700	.122	221	7.97	251	10.7	280	13.4						
1500	26460	.141	218	8.26	249	10.9	277	13.9	303	16.8				
1600	28230	.160	216	8.53	246	11.3	275	14.3	300	17.3	325	20.6		
1700	29990	.180	214	8.81	244	11.6	272	14.7	296	17.7	321	21.2	366	28.6
1800	31750	.202	212	9.18	241	12.0	268	15.0	294	18.2	318	21.7	363	29.0
1900	33520	.225	211	9.57	239	12.4	266	15.4	291	18.6	316	22.2	359	29.5
2000	35280	.250	210	10.1	238	12.8	265	15.7	288	19.1	312	22.6	356	30.1
2100	37050	.275	210	10.6	236	13.3	262	16.3	286	19.6	309	23.2	354	30.7
2200	38810	.302	209	11.3	236	13.8	260	16.8	284	20.1	307	23.7	351	31.3
2300	40580	.330	210	12.0	235	14.5	259	17.4	282	20.7	306	24.3	346	32.0
2400	42340	.360	211	12.8	235	15.1	258	18.2	280	21.4	302	24.9	345	32.7
2500	44100	.390	212	13.8	235	16.1	257	18.9	279	22.0	301	25.7	342	33.4
2600	45870	.422	214	14.6	236	17.0	256	19.7	278	22.9	299	26.4	340	34.1
2800	49400	.489	218	16.8	237	19.2	257	21.9	276	24.7	297	28.2	336	35.8
3000	52910	.560	224	19.4	241	21.7	259	24.3	278	27.1	296	30.3	335	37.9
3200	56450	.638	228	22.1	246	24.6	262	27.1	279	30.0	297	33.2	332	40.3
3400	59980	.721	236	25.5	251	27.7	267	30.4	282	33.2	248	36.2	331	43.1
3600	63510	.810	241	29.3	257	31.5	272	34.0	286	37.0	302	39.8	333	46.7
3800	67030	.900					277	38.1	291	40.9	305	44.1	335	50.6
4000	70560	1.000							296	45.5	309	48.4	336	55.2

Table XXVI.—No. 12 Niagara Conoidal Fan (Type N) Capacities and Static Pressures at 70°F. and 29.92 Inches Barometer

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	¼" S. P.		⅜" S. P.		½" S. P.		⅝" S. P.		¾" S. P.		⅞" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1000	21000	.063	97	1.50	121	2.35								
1100	23090	.076	96	1.67	119	2.51								
1200	25190	.090	97	1.90	119	2.69	139	3.72						
1300	27290	.106	98	2.16	118	2.94	138	3.93	156	5.07				
1400	29390	.122	100	2.49	118	3.24	137	4.21	154	5.31	172	6.64		
1500	31490	.141	103	2.87	119	3.63	136	4.51	153	5.62	170	6.88	186	8.31
1600	33600	.160	105	3.30	120	4.03	137	4.93	153	5.98	168	7.23	183	8.60
1700	35690	.180	108	3.80	123	4.52	138	5.39	152	6.41	168	7.63	182	9.02
1800	37790	.202	111	4.36	125	5.06	138	5.95	153	6.93	167	8.08	181	9.43
1900	39890	.225	114	4.94	128	5.67	140	6.55	153	7.52	167	8.66	180	9.95
2000	41990	.250	118	5.59	130	6.37	143	7.26	154	8.24	167	9.33	180	10.5
2100	44090	.275	121	6.27	133	7.16	145	8.01	156	8.96	168	10.1	180	11.3
2200	46190	.302	124	6.99	136	7.94	148	8.81	158	9.74	169	10.9	181	12.2
2300	48290	.330	128	7.83	139	8.84	150	9.74	161	10.7	171	11.8	182	12.9
2400	50390	.360	132	8.71	143	9.78	153	10.7	163	11.7	173	12.8	183	13.9
2500	52490	.390	136	9.62	146	10.7	156	11.8	165	12.7	175	13.8	185	15.0
2600	54590	.422	140	10.7	149	11.8	158	12.9	168	14.0	178	15.0	187	16.3
2800	58790	.489	148	13.0	156	14.2	165	15.4	173	16.6	183	17.6	192	18.7
3000	62980	.560	156	15.9	164	16.7	172	18.3	180	19.5	188	20.6	195	21.8
3200	67180	.638					179	21.3	187	22.8	195	24.1	203	25.4
3400	71380	.721									202	27.9	208	29.4

Outlet velocity, ft. per min.	Capacity, cu. ft. air per min.	Add for total press.	1" S. P.		1¼" S. P.		1½" S. P.		1¾" S. P.		2" S. P.		2½" S. P.	
			R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.	R.p.m.	Hp.
1300	27290	.106	205	9.22										
1400	29390	.122	203	9.49	230	12.7	257	16.0						
1500	31490	.141	200	9.84	228	13.0	254	16.6	278	20.0				
1600	33600	.160	198	10.2	226	13.5	252	17.0	275	20.6	298	24.5		
1700	35690	.180	196	10.5	223	13.7	249	17.4	272	21.0	294	25.2	336	34.0
1800	37790	.202	194	10.9	221	14.3	246	17.9	269	21.6	292	25.8	333	34.6
1900	39890	.225	193	11.4	219	14.7	244	18.3	267	22.2	289	26.4	329	35.1
2000	41990	.250	193	12.0	218	15.3	243	18.7	264	22.8	286	26.9	326	35.9
2100	44090	.275	193	12.6	217	15.8	240	19.5	263	23.3	283	27.7	324	36.6
2200	46190	.302	192	13.4	216	16.4	238	20.0	260	23.9	282	28.2	322	37.3
2300	48290	.330	193	14.3	215	17.3	238	20.7	258	24.6	280	29.0	318	38.0
2400	50390	.360	193	15.3	215	18.0	237	21.6	257	25.5	277	29.7	316	38.9
2500	52490	.390	194	16.4	215	19.2	236	22.5	256	26.2	276	30.5	313	39.8
2600	54590	.422	196	17.4	216	20.2	235	23.5	255	27.2	274	31.4	312	40.6
2800	58790	.489	200	20.0	218	22.9	236	26.1	253	29.4	273	33.6	308	42.6
3000	62980	.560	205	23.0	221	25.8	238	29.0	255	32.3	272	36.0	307	45.1
3200	67180	.638	209	26.4	225	29.2	240	32.3	256	35.7	273	39.5	304	48.0
3400	71380	.721	216	30.4	230	33.0	245	36.2	258	39.5	273	43.1	303	51.3
3600	75580	.810	221	34.9	236	37.5	249	40.5	263	44.1	277	47.4	305	55.6
3800	79780	.900					254	45.4	267	48.7	279	52.4	307	60.2
4000	83980	1.000							272	54.2	283	57.6	308	65.7

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